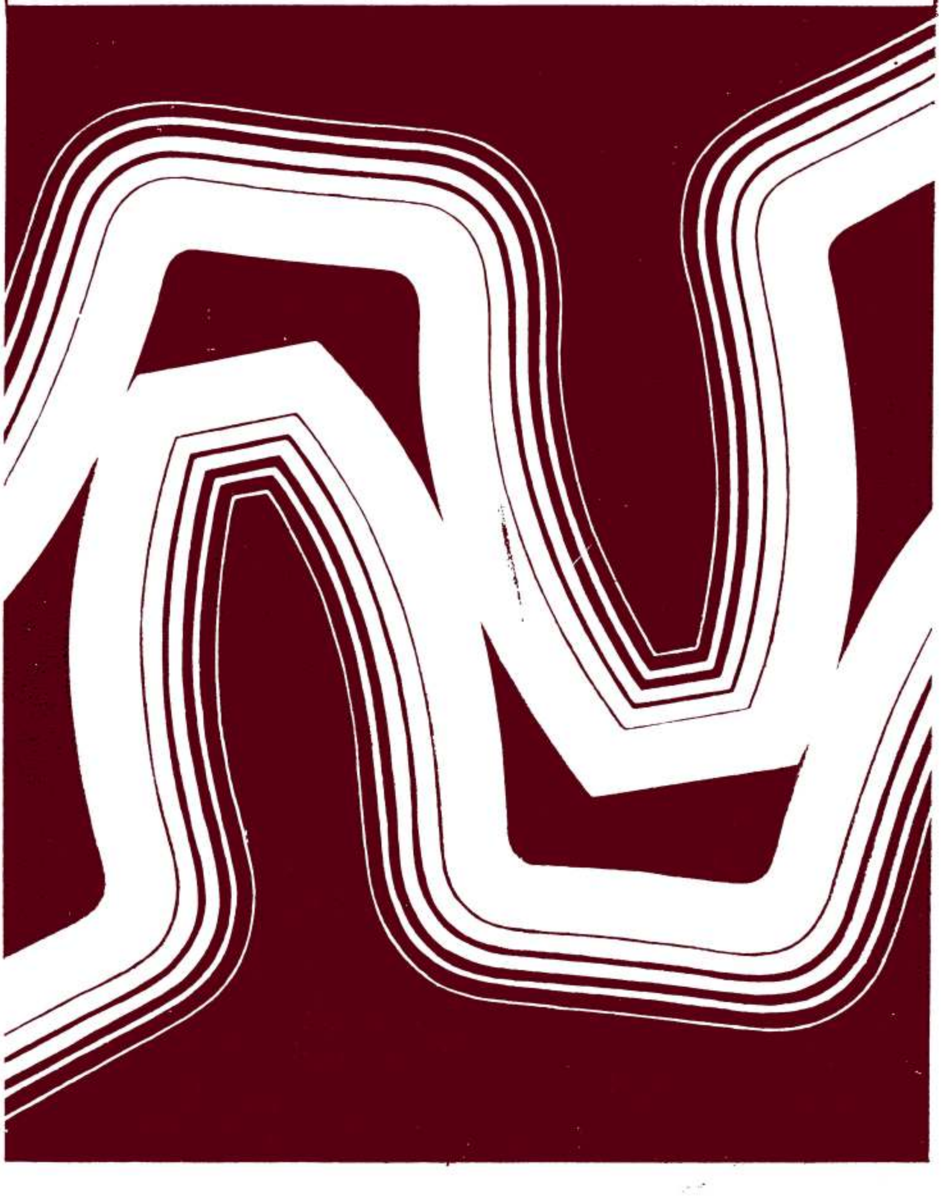
5

FUNDAMENTALS OF MACHINE DESIGN

MIR PUBLISHERS MOSCOW

P. ORLOV



This monograph is a valuable reference book for machine design engineers and students. It presents numerous constructions of machine-part elements in a systematized order. Advantages, disadvantages and typical applications of the designs are considered in sufficient detail to give the design engineer experience-proven guidelines for solving his specific problems. The book has been translated into Spanish. The first two volumes of the English translation were published in 1976 and the third and fourth, in 1977.

Mir Publishers of Moscow publish Soviet scientific and technical literature in eleven languages—English, German, French, Italian, Spanish, Czech, Serbo-Croat, Slovak, Hungarian, Mongolian, and Arabic. Titles include textbooks for universities, technical schools and vocational training, literature on the natural sciences and medicine, including textbooks for medical schools and schools for nurses, popular science, and science fiction. The contributors to Mir Publishers' list are leading Soviet scientists and engineers in all fields of science and technology and include more than 40 Members and Corresponding Members of the USSR Academy of Sciences. Skilled translators provide a high standard of translation from the original Russian. Many of the titles already issued by Mir Publishers have been adopted as textbooks and manuals at educational establishments in France, Cuba, Egypt, India and other countries.

Mir Publishers' books in foreign languages are exported by V/O "Mezhdunarodnaya Kniga" and can be purchased or ordered through booksellers in your country dealing with V/O "Mezhdunarodnaya Kniga".

5

FUNDAMENTALS OF MACHINE DESIGN

P. ORLOV

ИЗДАТЕЛЬСТВО • МИР •

П. И. ОРЛОВ

ОСНОВЫ КОНСТРУИРОВАНИЯ

ИЗДАТЕЛЬСТВО
«МАШИНОСТРОЕНИЕ»
МОСКВА

5 FUNDAMENTALS OF MACHINE DESIGN

P. ORLOV

TRANSLATED FROM THE RUSSIAN
BY F. PALKIN AND V. PALKIN

First published 1980

The Russian Alphabet and Transliteration

А а	a	К к	k	Х х	kh
Б б	b	Л л	l	Ц ц	ts
В в	v	М м	m	Ч ч	ch
Г г	g	Н н	n	Ш ш	sh
Д д	d	О о	o	Щ щ	shch
Е е	e	П п	p	Ъ	''
Ё ё	yo	Р р	r	Ы	y
Ж ж	zh	С с	s	Ь	'
З з	z	Т т	t	Э э	e
И и	i	У у	u	Ю ю	yu
Й й	y	Ф ф	f	Я я	ya

The Greek Alphabet

Α α	Alpha	Ι ι	Iota	Ρ ρ	Rho
Β β	Beta	Κ ϰ	Kappa	Σ σ	Sigma
Γ γ	Gamma	Λ λ	Lambda	Τ τ	Tau
Δ δ	Delta	Μ μ	Mu	Υ υ	Upsilon
Ε ε	Epsilon	Ν ν	Nu	Φ φ	Phi
Ζ ζ	Zeta	Ξ ξ	Xi	Χ χ	Chi
Η η	Eta	Ο ο	Omicron	Ψ ψ	Psi
Θ θ ϑ	Theta	Π π	Pi	Ω ω	Omega

На английском языке

Contents

Chapter 1

Fastenings

1.1. Types of Threaded Fastenings

Three main types of threaded fastenings are used in mechanical engineering: (1) bolts with nuts (Fig. 1, *I*); (2) cap screws (Fig. 1, *II*, *III*); and studs (Fig. 1, *IV*, *V*).

(1) **Fastening with bolts and nuts** is used only where assembled parts can be provided with through bolt holes. Such parts are not very convenient to assemble, because the bolt must be fixed against rotation when the nut is torqued. It is also desirable that the bolt

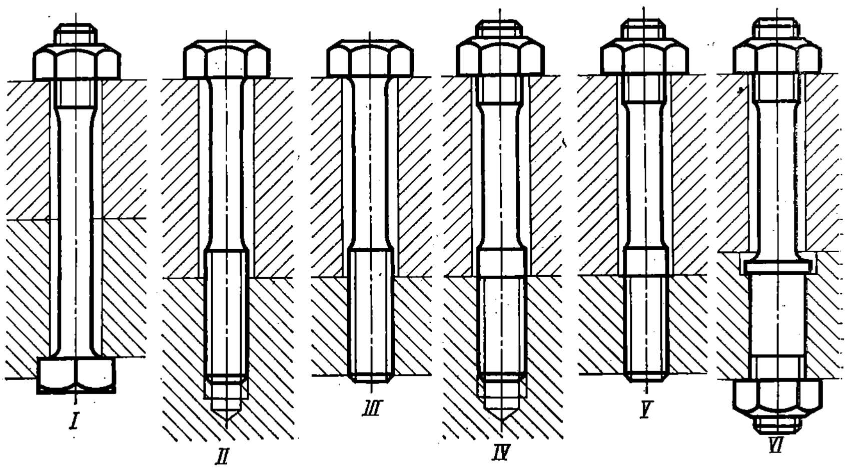


Fig. 1. Main types of screw fastenings

should be fixed axially during nut tightening. Two elements are manipulated during assembly: the bolt and the nut.

(2) **Fastening with cap screws** is used for parts with blind threaded holes (Fig. 1, *II*), for assemblies where bolts with nuts are impossible to employ, or for parts with through tapped holes (Fig. 1, *III*), where screws can be set only from one side of the joint. Cap screws are similar in design to bolts.

Parts with threaded holes should be made of materials which provide strong screw threads (steel, malleable and high-strength cast irons, titanium alloys, bronze, etc.). Where soft alloys (based on aluminium, magnesium, zinc, etc.) are used, threaded inserts of tougher materials should be introduced in the parts.

Threaded holes in grey cast iron are undesirable for frequently disassembled parts (screw threads in grey iron tend to crumble and quickly wear out) and for stainless-steel parts (where screw threads are difficult to tap because of the material's toughness). When its thread becomes worn out, a threaded part can be repaired (wherever possible) only by introducing therein a threaded insert.

The height of parts assembled with cap screws is somewhat limited; long cap screws are difficult to tighten properly, because the screw's shank is inevitably twisted during tightening. Unlike assembly with the use of bolts with nuts, that with screws is more advantageous, because a single fastening element, the screw, is manipulated here.

(3) **Fastening with studs** (Fig. 1, *IV*, *V*) is mainly used for parts made of soft or brittle materials (aluminium and magnesium alloys, grey iron), and also for parts with blind or through threaded holes where cap screws are impossible to mount.

A stud is tightly screwed into a tapped hole, often by an interference fit. This provides a fairly reliable screw joint even in parts of soft materials.

A part with tapped holes becomes inoperable when the thread is stripped out or otherwise damaged; but it can be restored by using a suitable threaded insert. When a stud breaks away, its metal end is difficult to remove from the tapped hole.

Assembly and disassembly is specific here: the assembled parts are put together by being moved only in the direction perpendicular to the joint face; and for disassembly the part being removed must be lifted all the way up to clear the projecting ends of the studs. That is different from fastening with bolts and cap screws, where the assembled parts can be displaced along the joint face after the fasteners have been removed. A single fastening element, the nut, is manipulated for assembly and disassembly.

Fastening with studs is somewhat cumbersome for assembly, since the projecting ends of the studs hamper access to adjacent parts. It is especially so in multi-stud assemblies with a forest of long studs.

In addition to the above-mentioned types of fastenings, use is made of various combination types. An example is the one shown in Fig. 1, *VI*, which is used fairly often. A special bolt is set in a plain through hole of one of the assembled parts and is fastened with a nut; the other part is fastened with another nut screwed on the other end of the bolt. The bolt is secured in the plain hole by the nut, which

makes it similar to fastening with bolts; the permanent position of the bolt in one part and its threaded end used to fasten with a nut the other part make it similar to fastening with studs.

With relation to design, cap screws and studs are more advantageous than bolts because they make it possible to develop the shape of assembled parts with greater freedom.

Fastening with bolts (Fig. 2, *I*) requires that the mating parts should have flanges, which impose dimensional limitations and determine the outline of the parts.

Fastening with studs and cap screws allows the part shape to be diversified. With studs, flanges can be retained (Fig. 2, *II*), or one of the assembled parts can be altered so that with the same location of the fasteners it becomes larger and stiffer (Fig. 2, *III-VI*).

With cap screws, similar options are available.

General-purpose screw fasteners are usually made of grade 45 steel, whereas heavy-duty ones (connection-rod bolts, heavily loaded studs,

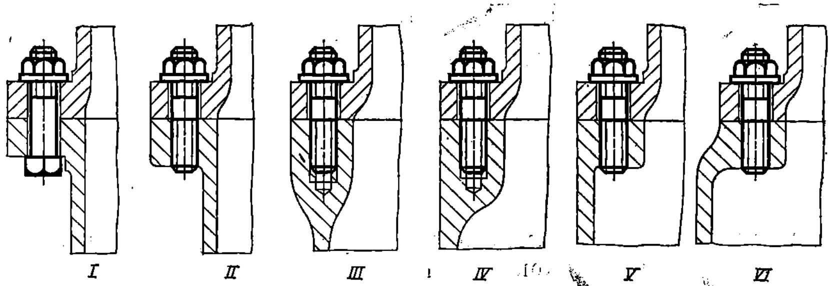


Fig. 2. Variations in the shape of a housing-type part allowed by fastening with studs and cap screws

etc.), are made of chromium steels, grades 40X and 30XГC, high-temperature steels, grades 30XM, 50XФA, 25X12M1Ф, and stainless steels, grades 30X13, 40X13.

Studs with the shank equal in diameter to the thread are normally made from bars of ground steel (also known as silver steel), which has high diametral accuracy and good surface finish. The plain portion of the shank is left unmachined. Bolts are cold or hot headed, depending on the grade of steel they are made of, soft or hard.

In mass and batch production, screw thread on fasteners is formed by rotating tool heads and thread milling cutters. However, the most productive method, which at the same time provides the strongest thread, is thread rolling.

Bolts and studs with shanks of reduced diameter are turned from round blanks. The shanks of heavy-duty fasteners are reduced in diameter by swaging in rotary die presses.

1.2. Nuts and Bolt Heads

Hexagon nuts and bolt heads have found the greatest use. Square nuts and bolt heads, and also round nuts with two flats, are used less frequently. In some instances (for large thread diameters) octagon nuts can be found.

Width Across Flats. The principal dimension for hexagon and square nuts and bolt heads, and also for round nuts and bolt heads with two flats, is the distance between parallel flats, termed "width across flats" S. Figure 3, *I* shows a round nut (or bolt head) with two flats. The minimum diameter of the cylinder to ensure reliable grip of the nut or bolt head by a spanner should be $D_{min} = (1.1 \text{ to } 1.2)\ S$. The

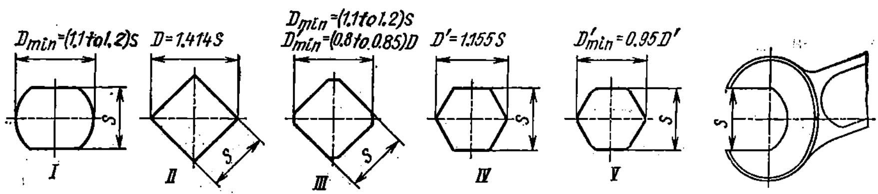


Fig. 3. Determining width-across-flats

maximum diameter is limited by the length of the jaws of open-ended spanners, and it is taken to be $D_{max} = 1.4S$.

For square nuts or bolt heads (Fig. 3, *II*), the width across corners is $D = 1.414\ S$.

For externally rounded square nuts or bolt heads (Fig. 3, *III*), the minimum diameter to ensure reliable grip by a spanner should be $D_{min} = (1.1 \text{ to } 1.2)\ S$, or $D_{min} = (0.8 \text{ to } 0.85)\ D$.

For hexagon nuts or bolt heads (Fig. 3, *IV*) the width across corners is $D' = 1.155\ S$, and for externally rounded hexagon nuts or bolt heads (Fig. 3, *V*) the minimum width across corners to ensure reliable grip by a spanner is $D'_{min} = 1.1\ S$, or $D'_{min} = 0.95\ D'$.

The standard values of width across flats, S (in mm), are as follows: 3; 3.2; 3.5; 4; 4.5; 5; 5.6; 6; 7; 8; 9; 10; 11; 12; 14; 17; 19; 22; 24; 27; 30; 32; 36; 41; 46; 50; 55; 60; 65; 70; 75; 80; 85; 90; 95; 100; 110; 119; 120; 125; 130; 135; 140; 145; 150; 155; 160; 165; 170; 175; 180; 185; 190; 200; 210; 220; 230; 240; 250.

The width across flats, S, for general-purpose fasteners is held to within the C_5 tolerance grade, and for high-accuracy fasteners the C_3 or C_{3a} tolerance grades. The wrench openings are held to within the X_3 or X_5 tolerance grades, depending on the required accuracy.

1.3. Dimensional Relationships

Three types of hexagon nuts and bolt heads are distinguished:
(1) light nuts and bolt heads (Fig. 4, *I*);
(2) regular nuts and bolt heads (Fig. 4, *II*);
(3) heavy nuts and bolt heads (Fig. 4, *III*).

Figure 5 shows average values of the width across flats (S) and across corners (D), and also values of S/d and D/d as a function of thread diameter d for bolt heads of the light type (*I*), regular type (*II*), and heavy type (*III*).

There is a trend in engineering to the use of bolt heads and nuts of the light type; being strong enough, they have smaller radial dimensions and mass, which makes it possible to design compact screw joints.

The thickness h of nuts* (Fig. 6) is taken to be (0.6 to 1.2)d, where d is the nominal thread diameter.

Experience has shown that a nut with thickness $h = 0.7d$ is equal in strength to the threaded part of the mating bolt. Where $h > 0.7\ d$, it is always the bolt shank under the nut that is broken off under tensile loads. The height of bolt heads (Fig. 7) is made equal to (0.6 to 0.8) d.

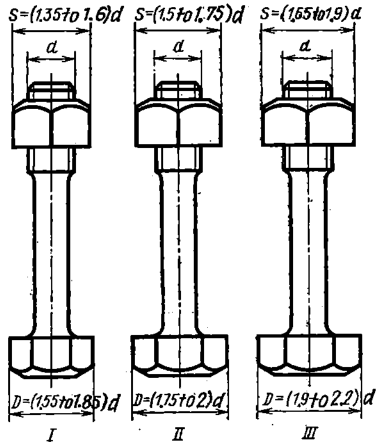


Fig. 4. Dimensions of hexagon nuts and bolt heads

The hexagon's end faces are chamfered at an angle of 120°. The chamfers facilitate engagement of the hexagon by a spanner; in nuts, the chamfers also provide annular bearing surfaces. The size of the chamfers depends on diameter d_1 of the end face (Figs. 8 and 9). This diameter, formerly $d_1 = S$ (S = width across flats), has now been standardized for nuts at $d_1 = (0.9 \text{ to } 0.95)\ S$.

For bolt heads, d_1 can be reduced to (0.7 to 0.6) S (Fig. 9) without detriment to their strength, which helps to reduce the mass of assemblies. Reduction in diameter d_1 for nuts results in the smaller bearing surfaces.

When presented in a side view, hexagons with a reduced diameter d_1 differ from those with $d_1 = S$. The circular arcs that graphically represent the hyperbolic lines of intersection of conical chamfers with flats are not tangent (as is the case with $d_1 = S$) to the lines of the hexagon end faces (see Fig. 8).

* According to the relevant USSR Standards, nuts are made (1) thin, with $h = 0.6d$, where d is the thread diameter; (2) regular, $h = 0.8d$; (3) thick, $h = 1.2d$; and (4) extra thick, $h = 1.5d$.

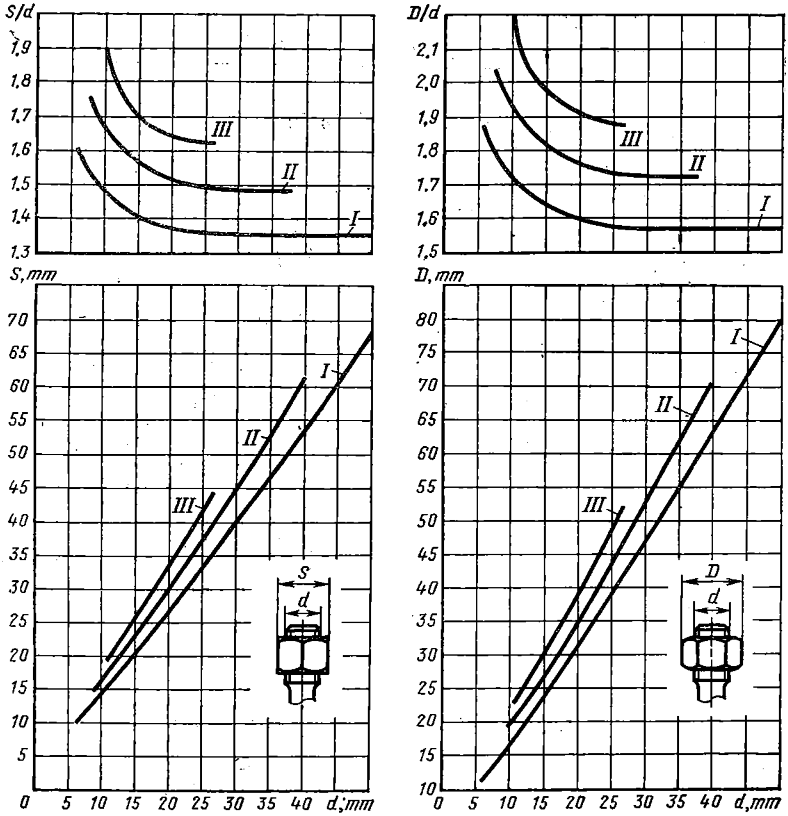


Fig. 5. Width-across-flats S and width-across-corners D for hexagon nuts and bolt heads as related to thread diameter d

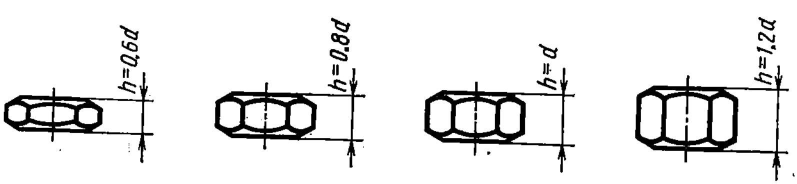


Fig. 6. Height of hexagon nuts

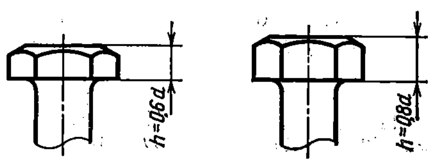


Fig. 7. Height of hexagon bolt heads

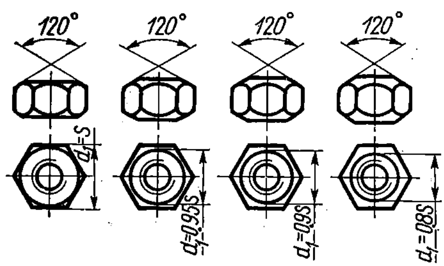


Fig. 8. Chamfers on hexagon nuts

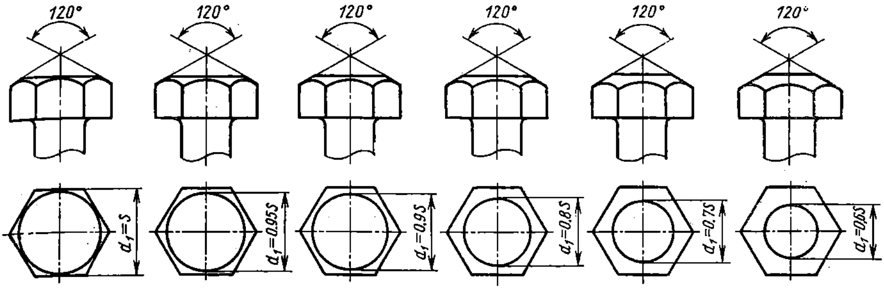


Fig. 9. Chamfers on hexagon bolt heads

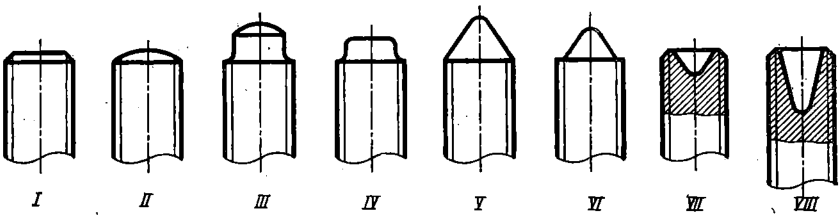


Fig. 10. Ends of fasteners' threaded shanks

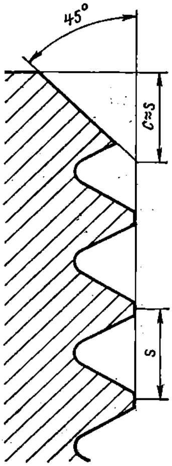


Fig. 11. Chamfer on the threaded shank of fasteners

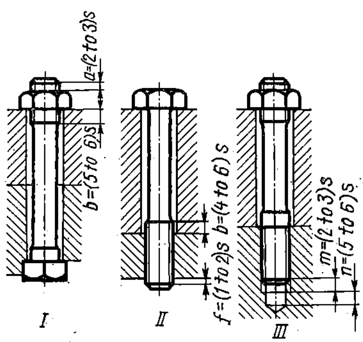


Fig. 12. Amount of unengaged thread for the main types of threaded assemblies

This difference is often ignored in drafting practices, where, for simplicity, hexagons are usually drawn as if $d_1 = S$.

Figure 10 shows the ends of the threaded shanks of fasteners. The most common is the form of Fig. 10, *I* with a 45° chamfer. The height *c* of the chamfer is usually taken to be equal to the thread pitch *s* (Fig. 11).

A spherical end (Fig. 10, *II* and *III*) is used for pressure screws, a rounded cylindrical end (Fig. 10, *IV*) for large bolts and screws, and conical ends (Fig. 10, *V* and *VI*) for guiding the screws into difficult-to-access tapped holes. Recesses in bolt ends (Fig. 10, *VII* and *VIII*) are provided for better compliance of the end threads.

The lengths of unengaged thread for the main types of threaded fasteners are shown in Fig. 12.

Length *a* of unengaged thread on a bolt above a nut (Fig. 12, *I*) should be equal to 2-3 thread pitches; length *b* under the nut or above a tapped hole (Fig. 12, *II*) should be no less than 4-6 pitches; the end of a cap screw or stud should extend beyond the tapped hole (Fig. 12, *II*) by amount *f* equal to 1-2 pitches.

The given requirements apply to standard medium-size bolts, screws and studs. For long fasteners, these amounts should be determined by calculating the dimension chain of a specific assembly.

Figure 12, *III* illustrates the requirements for assemblies where studs (or cap screws) are put into blind threaded holes. Length *m* of unengaged full-form thread under the end of a stud should be no less than 2-3 pitches; distance *n* from the last full-form thread to the bottom of a hand-tapped hole should be no less than 5-6 pitches. For machine-tapped holes, distance *n* should be made twice as long.

1.4. Distribution of Load Among Threads

In conventional screw joints, the applied load is distributed nonuniformly. The threads nearest to the nut bearing surface are loaded much heavier than the successive threads. Theoretical analysis and experimental data reveal that the first thread takes up to 30% of the load, whereas the threads farthest from the nut bearing surface practically remain unloaded. The reason for this phenomenon is dissimilar deformations of the nut and bolt threads under load. The regions of the bolt shank nearest to the nut bearing surface are stretched by the full load. The threads deflect together with the shank in the direction of application of the load (Fig. 13).

In the nut, the situation is quite reverse, i.e., the regions of the nut body nearest to its bearing surface are compressed by the full load, and the nut threads deflect in the direction opposite to that of the deflection of the shank threads. The first thread has the greatest deformation and hence takes most of the load. The following threads of the shank are stressed and deformed to a lesser degree. Accordingly,

compressive strain in the nut decreases, and so does the load on each following thread. This effect is stronger with greater absolute values of tensile strain in the bolt and compressive strain in the nut, i.e. with higher stresses in the screw joint. Therefore, it is expedient to

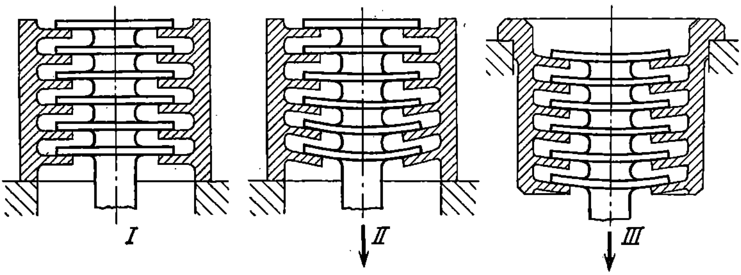


Fig. 13. Diagram of load distribution among screw-joint threads
I and *II* — compression nut before and after loading, respectively; *III* — tension nut after loading

increase the diameter of thread on bolts and in nuts where greater strength and more uniform load distribution must be achieved (Fig. 14).

Nonuniform load distribution takes place in assemblies with cap screws and studs, where the cross-sections of the parts with threaded

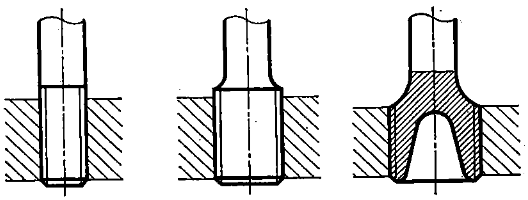


Fig. 14. Increasing thread diameter for greater uniformity of load distribution among the threads

holes are much greater than those of the fasteners. An exception is parts of light alloys with a low value of Young's modulus, which reduces their stiffness.

Screw joints are to some extent capable of self-strengthening. If stresses in the most heavily loaded threads exceed the yield point, plastic deformations (shearing and crushing) of these threads will take place and cause an increase in pitch in the most heavily loaded threads of the nut and decrease in pitch in those of the bolt. As a result, the load per thread will level off along the screw joint. This effect is typical of screw joints of soft and plastic materials. Threaded parts of hard and strong materials are capable of self-strengthening to a much lesser degree.

A number of effective methods can be used to provide a uniform distribution of load among the threads of the nut and bolt.

One method is to make the nut and the bolt deflect in the same direction. This can be achieved by arranging the bearing surface on the nut above its threaded portion (Fig. 15, *II*). The nut portion under the bearing surface ('skirt') is stretched; the threads of the nut deflect in the same direction as do the threads of the bolt, resulting in a more uniform load distribution among the threads.

Such nuts, which can be termed *tension nuts* (as distinct from conventional nuts termed *compression nuts*), are widely used in critical applications. The drawbacks to this type of nut are its greater axial

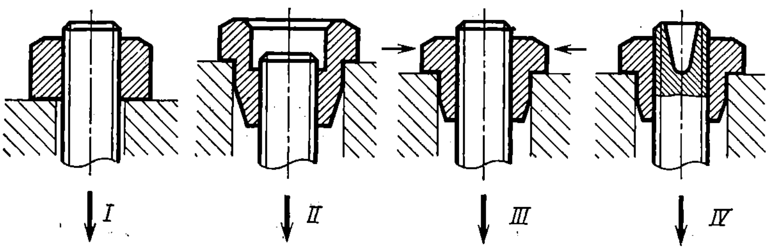


Fig. 15. Nuts
I — compression type; *II* — tension type; *III* — combination (tension-compression) type; *IV* — same, with recessed end of bolt shank

and radial dimensions, and also the need for increased diameter of the bolt hole to accommodate the nut skirt.

If the bearing surface of a nut is disposed in a plane between the first and the last thread (Fig. 15, *III*), the nut is of the *tension-compression* type. The nut body below the bearing surface is subject to tension, and above the bearing surface, to compression. Being smaller in size than tension nuts, nuts of the latter type perform equally well, because here the positive effect is produced both by elongation of the skirt and by contraction of the upper part of the nut, which tightens on the upper threads of the bolt.

Still greater uniformity of load distribution is achieved when a recess is provided on the end face of the bolt shank (Fig. 15, *IV*), which increases the compliance of the upper threads. This method is used mainly for large-diameter bolts.

Another method for attaining a uniform load distribution is to use a nut with a thread pitch a few microns greater than that of the mating bolt (Fig. 16). The joint thus formed operates as schematically shown in Fig. 17. In the initial no-load position (Fig. 17, *I*), there is some clearance between the lower threads of the bolt and of the nut. As a load is applied to the bolt, it is stretched and the nut is compressed, and the bolt threads successively come into contact with the nut threads (Fig. 17, *II*). When fully loaded, all the threads bear

the load uniformly (Fig. 17, *III*). This method does not require special-shape nuts; taps with a slightly increased pitch of thread is all that is needed.

The necessary increase of the nut thread pitch can be determined as follows.

Suppose the bolt shank is loaded by force P (Fig. 18). The clearance between the lowermost threads of the bolt and of the nut is $h = z\Delta s$, where Δs is the increase in the pitch of the nut thread over that of the bolt thread, and z is the number of the preceding threads (counting from the first thread of the nut that is in tight contact with the associated thread of the bolt).

This clearance h must be taken up by elongation λ_{el} of the bolt shank and shortening λ_{sh} of the nut due to force P, i.e., the condition $h = \lambda_{el} + \lambda_{sh}$ must be met.

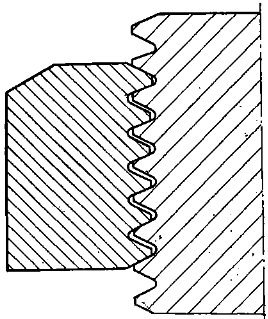


Fig. 16. Thread profile in screw joints with increased-pitch nuts

The elongation of the bolt shank

$$\lambda_{el} = \frac{Pl}{EF_1}$$

The shortening of the nut

$$\lambda_{sh} = \frac{Pl}{EF_2}$$

where $l = zs$ = length of thread engagement; E = modulus of elasticity (the nut and bolt are presumed to be made of the same

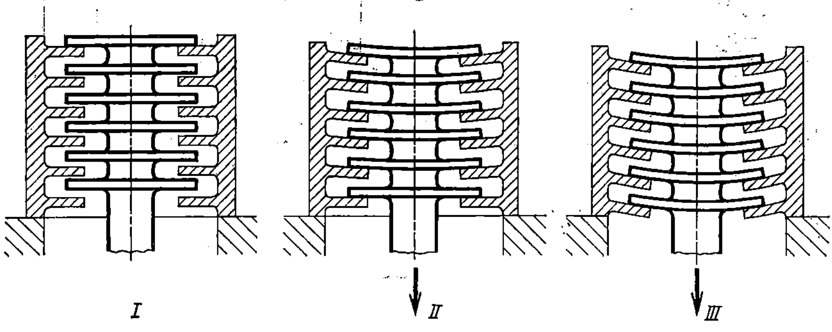


Fig. 17. Diagram of load distribution among threads of screw joints with increased-pitch nuts

I — before loading; *II* — initial phase of loading; *III* — after loading

material); F_1 = cross-sectional area of bolt threaded part; F_2 = = cross-sectional area of nut.

Then

$$h = z\Delta s = \frac{Pzs}{E}\left(\frac{1}{F_1} + \frac{1}{F_2}\right)$$

Hence,

$$\frac{\Delta s}{s}=\frac{P}{E}\left(\frac{1}{F_1}+\frac{1}{F_2}\right)=\frac{P}{EF_1}\left(1+\frac{F_1}{F_2}\right)$$
$$=\frac{\sigma_1}{E}\left(1+\frac{F_1}{F_2}\right) \tag{1}$$

where σ_1 = stress in bolt threaded part.

The bolt and nut cross sections can be approximately expressed as

$$F_1=\frac{\pi d_0^2}{4} \quad \text{and} \quad F_2=\frac{\pi}{4}(S^2-d_0^2)$$

where d_0 = pitch diameter of thread; S = width across flats of the nut.

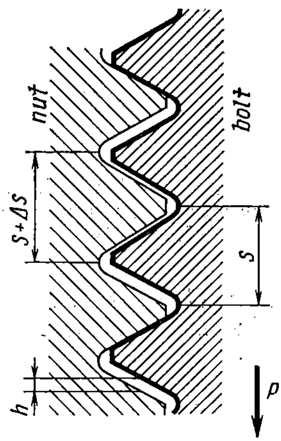


Fig. 18. Determining difference in pitch between nut and bolt threads

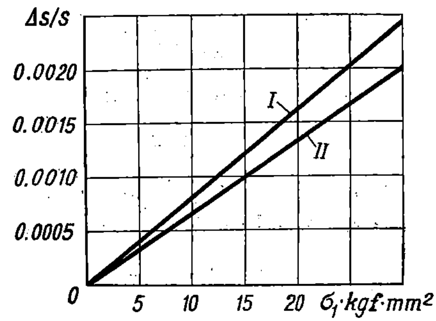


Fig. 19. Relative pitch difference $\Delta s/s$ as a function of stress σ_1 of bolt shank

I — light nuts; *II* — regular nuts

Then

$$\frac{\Delta s}{s}=\frac{\sigma_1}{E}\,\frac{1}{1-\left(\frac{d_0}{S}\right)^2}$$

Within the range of the most commonly used thread diameters (d = 8 to 20 mm), ratio $\frac{d_0}{S}\approx 0.67$ for light nuts and $\frac{d_0}{S}\approx 0.6$ for regular nuts.

Hence,

$$\frac{\Delta s}{s}=(1.8 \text{ to } 1.5)\frac{\sigma_1}{E} \tag{2}$$

where 1.8 is used for light nuts, and 1.5 for regular nuts.

The graph in Fig. 19 shows the relationship between tensile stress σ_1 in the bolt shank and the values of $\frac{\Delta s}{s}$ for light and regular nuts, as found from formula (2). The value of E in the formula is taken at 22×10^5 kgf/cm^2 (for steel).

With $\sigma_1 = 2{,}000$ kgf/cm^2 in fasteners for high-load applications, $\frac{\Delta s}{s} = 0.0016$ for light nuts. Therefore, the nominal thread pitch being 2 mm, for instance, the nut thread should have a pitch greater than that of the bolt thread by $0.0016 \times 2 = 3.2$ µm.

It can be seen from the graph of Fig. 19 that the required difference in the nut and bolt thread pitches depends on the stress in the bolt shank. Consequently, in a screw joint with the difference in the pitches found from the design value of stress, the upper threads come into action at the beginning of loading. As the load rises to the rated value, all the lower flanks of the bolt thread get in contact with the upper flanks of the nut thread, which results in a uniform distribution of the load among the threads.

If the rated load is exceeded, the lower threads will prove to be loaded heavier.

The relationship between the pitch difference and the stress indirectly supports the conclusion that it is expedient to increase as much as possible the bolt and nut cross-sections, i.e. thread diameter.

Some other methods for increasing the uniformity of load distribution in a thread are the use of nuts with a slightly tapered thread (Figs. 20, 21), nuts with conically truncated lower ridges of the thread (Figs. 22, 23), nuts with a special-profile thread providing for increased compliance of individual threads (Fig. 24), and thread coatings. But they are less effec-

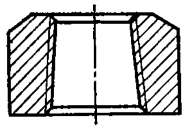


Fig. 20. Nut with conical thread

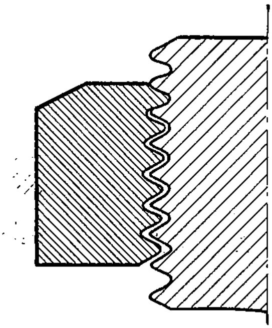


Fig. 21. Thread profile in screw joints with conical-thread nut

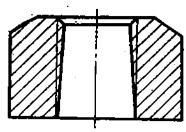


Fig. 22. Nut with conically truncated thread

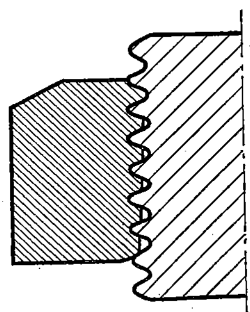


Fig. 23. Thread profile in screw joints with nut having conically truncated thread

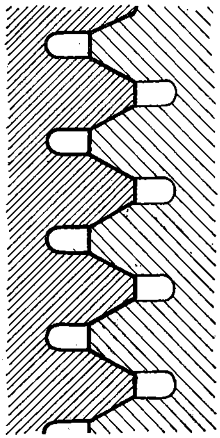


Fig. 24. Profile of high-compliance thread

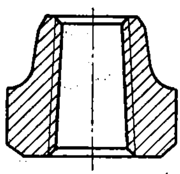


Fig. 25. Nut with diametrically squeezed upper part

tive and more difficult to carry out in production. Thus, for instance, a tapered thread cannot be cut by tapper taps, although this high-output thread cutting method is indispensable to mass production of nuts. A truncated thread profile requires additional machining operations.

Fig. 25 shows a nut with a radially squeezed upper part, so that it is similar to nuts with a tapered thread.

Greater uniformity of load distribution can also be achieved by introducing an elastic layer between the threads of the bolt and nut. Such a layer is obtained, for instance, by zinc or cadmium plating of the threads. The drawback of this method is that the layer gradually wears out, especially in assemblies which are frequently dismantled.

Fig. 26 illustrates other methods for increasing the uniformity of load distribution in a thread. Nuts with recesses at the bottom threads are shown in Fig. 26, *I-VII*. Such recesses serve a double purpose: to increase compliance of the bottom threads and to provide diametrical contraction of the upper threads by the forces acting on the bearing surface of

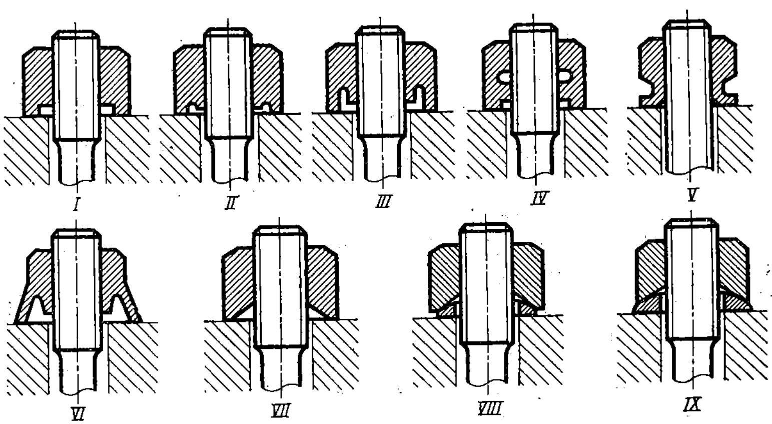


Fig. 26. Nuts with recesses at bottom threads

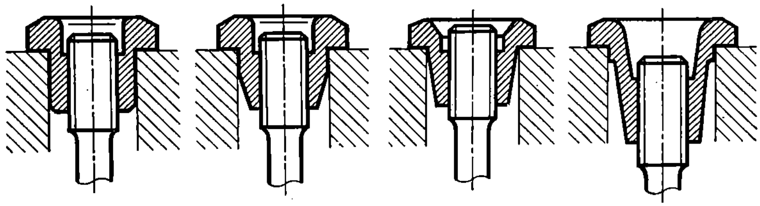


Fig. 27. Tension nuts

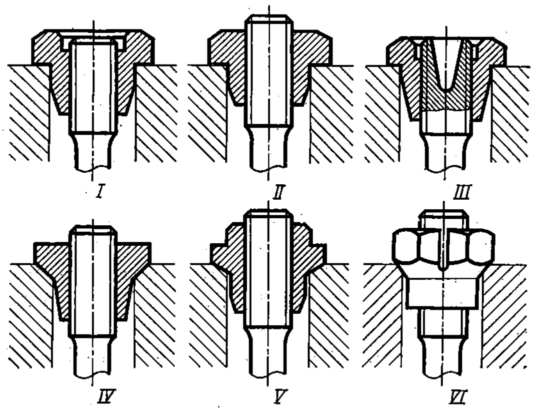


Fig. 28. Nuts of tension-compression type

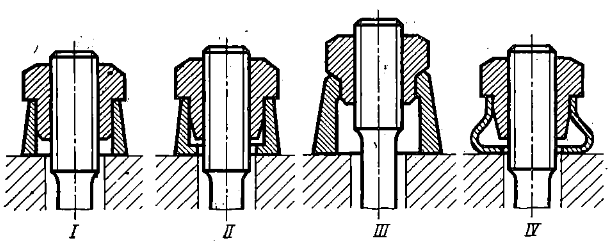


Fig. 29. Mounting nuts on spacers

the nut as it is tightened. The latter effect is especially typical of the nut shown in Fig. 26, *IV*. In the nuts of Fig. 26, *VIII*, *IX* the same effect is achieved by placing spherical seat washers under the nuts; the washers also make for self-alignment of the nuts.

Various tension nuts are shown in Fig. 27, and tension-compression nuts in Fig. 28, *I-VI*. In the nuts of Fig. 28, *IV*, *V*, the bearing surface is tapered for better contraction of the upper threads. This effect is further increased by slots in the upper part of the nut body.

A drawback common to tension nuts—the need for increased diameter of bolt holes to receive the skirt of a nut (sometimes with chamfering the holes as shown in Fig. 28, *IV-VI*)—is remedied by mounting the nuts on spacers (Fig. 29, *I-III*). Fig. 29, *IV* shows a spring spacer which also serves to lock the nut.

1.5. Bolts

Standard bolts are classified by accuracy (and finish) as (1) unfinished, (2) semi-finished and (3) finished.

Coarse and fine threads are used for standard bolts. When selecting fasteners, priority should be given to those with coarse threads.

GOST 1759-70 establishes grades of strength for bolts, screws, and studs made of carbon and alloy steels. Each grade is designated by two numbers, the grades being 3.6; 4.6; 4.8; 5.6; 5.8; 6.6; 6.8; 6.9; 8.8; 10.9; 12.9; 14.9. The first number multiplied by 10 determines the lower limit of ultimate strength in kgf/mm^2, and the second number multiplied by 10 determines the ratio of the yield point to the ultimate strength in %; the product of the numbers determines the yield point in kgf/mm^2. For nuts made of the same materials, the following grades of strength are established: 4, 5, 6, 8, 10, 12, 14. Each number multiplied by 10 gives the value of load-induced stress in kgf/mm^2.

For bolts, screws, and studs of stainless steels, high-temperature steels, and the like, the following groups determining their mechanical properties are established: 21, 22, 23, 24, 25, 26. The mechanical properties of nuts are determined by groups 21, 23, 25, 26.

The main types of bolts are presented in Fig. 30.

Fig. 30, *I* shows a plain bolt whose shank is equal in diameter to its thread. This form is used for short bolts or bolts for light loads. For critical applications, use is made of 'elastic' bolts (Fig. 30, *II*) with diameter d_0 of the shank reduced to the thread minor diameter, or even to 0.8-0.7 of the thread nominal diameter.

As is known, increased elasticity of bolts improves operating conditions for screw joints which are subjected to impact loads.

Bolts with a shank of reduced diameter are not so much affected by misalignment caused by out-of-squareness of the bearing surfaces of the bolt head and the nut and by non-parallelism of the thread to the bolt axis. A shank of reduced diameter makes it possible to introduce smoothly curved junctions between the shank and head and also between the shank and the threaded part of the bolt, thereby improving its fatigue strength. This increases resistance of elastic bolts to cyclic loads.

Some bolt varieties feature centring collars on the shank near the head and the thread (Fig. 30, *II-IV*). It is often, however, that such

collars are disposed of (Fig. 30, *V*) to increase elasticity of the bolt and allow its self-alignment in the bolt hole.

Hexagon bolt heads are normally used (Fig. 31, *I*, *II*), although heads of some other shapes can also be found in practice: round heads

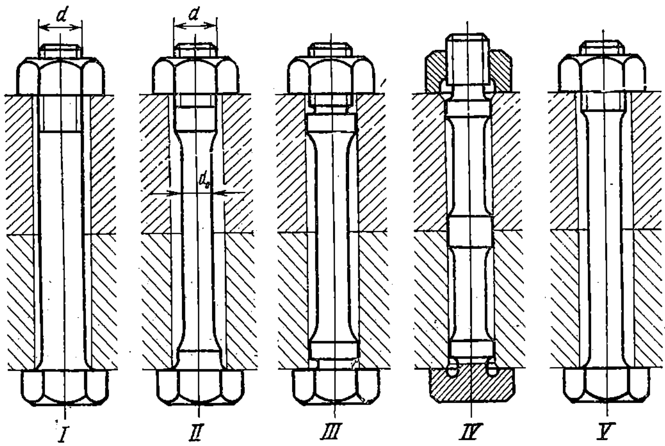


Fig. 30. Main types of bolts

with flats (Fig. 31, *III*, *IV*), hexagon socket heads (Fig. 31, *V*), serrated round heads (Fig. 31, *VI*), etc.

Hexagon socket heads find the most frequent use for bolts placed in counterbored holes (Fig. 31, *VII*), where the bolt head cannot be engaged by a plain open spanner.

The strength of bolts depends considerably on the contour of the junction between the shank and the head and the threaded part of

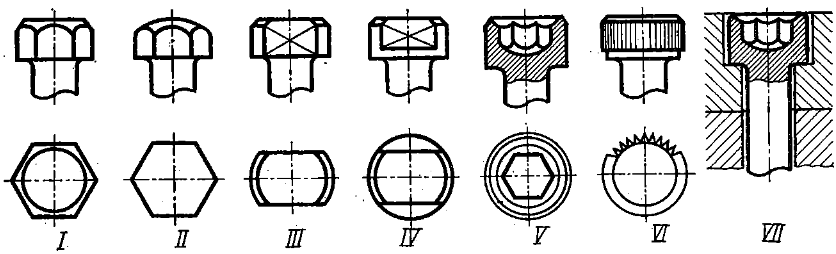


Fig. 31. Shapes of bolt heads

the bolt. There should be a fillet with radius *R* of no less than $0.2d$ between the head and the shank (Fig. 32, *II*). Fig. 32, *I* shows a bolt without any fillet.

The reducing of bolt shanks provides the possibilities of using fillets most favourably contoured for increased fatigue strength, e.g.,

conical (Fig. 32, *III*) and elliptical (Fig. 32, *IV*), and also relief recesses (Fig. 32, *V-VII*). The most advantageous in this respect are bolt heads with tapered bearing surfaces (Fig. 32, *VIII*, *IX*).

Fillets between the threaded and the plain portions of bolt shanks (Fig. 33, *I*) should have a radius R of no less than $(d - d_0)/2$ (where

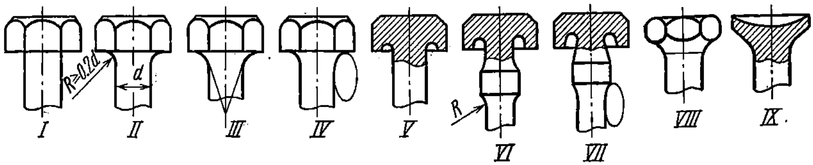


Fig. 32. Junction of bolt heads with shanks

d is the major diameter of thread, and d_0 is the shank diameter), preferably R is taken equal to d (Fig. 33, *II*), or they should be conical (Fig. 33, *III*) or elliptical (Fig. 33, *IV*), which provides a steep

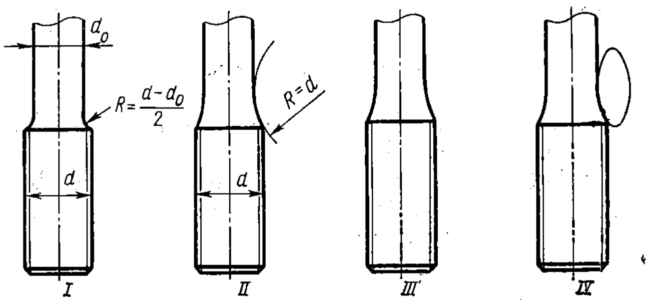


Fig. 33. Junction of plain and threaded parts of bolt shanks

end of the thread and its smooth junction with the shank plain portion.

Fillets of the same types are introduced between the centring collars (Fig. 32, *VI*, *VII*) and the reduced portions of bolt shanks.

When bolts, even those with a shank of reduced diameter (Fig. 30, *V*), are not elastic enough to ensure proper functioning of the screw joint, use is made of special elements introduced for added elasticity (Fig. 34). Where axial dimensions are not critical, the length of the bolt is increased, and a bushing (Fig. 34, *III*) or a spring element (Fig. 34, *IV*) is placed under the nut. If axial dimensions are limited, elastic elements are extended radially, e.g. use is made of elastic washers (Fig. 34, *V*, *VI*).

Fig. 34, *VII* shows a peculiar design which provides high elasticity with small axial and radial dimensions of the assembly. The bolt is mounted in two concentric sleeves; as the nut is tightened, the

outer sleeve *a* is stretched and the inner sleeve *b* is compressed. The cross-sectional areas of the bolt and the sleeves are equal. Consequently, the joint is three times as elastic as the bolt as such.

It is necessary for the proper functioning of a screw joint that the force applied to it should act along its axis, or, in other words, that the joint should have no misalignments and the bolt should be free of bending. Elastic bolts themselves compensate for misalignments

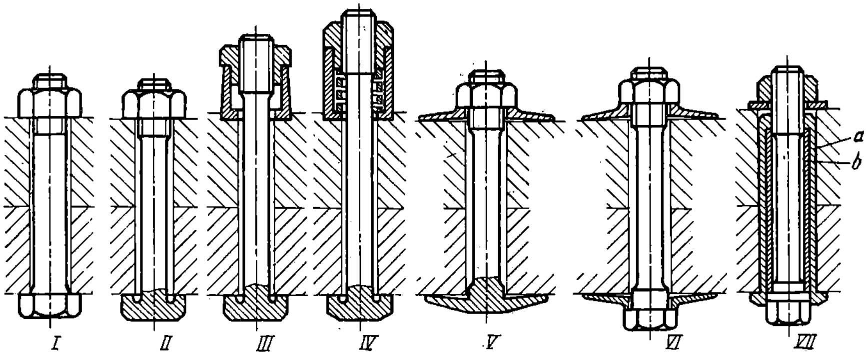


Fig. 34. Methods for increasing elasticity of bolt assemblies

I—rigid bolt; *II*—elastic bolt; *III*—long elastic bolt; *IV*—bolt with helical spring; *V*—bolt with elastic head and spring washer; *VI*—bolt with spring washers; *VII*—bolt with tension (*a*) and compression (*b*) sleeves

fairly well, but bending gives rise to undesirable stresses in the bolt shank. For this reason, special measures are taken to prevent misalignments, e.g. a somewhat loosely fitting thread in the joint is used. Contrary to the formerly common beliefs that the thread should be made as close-fitting as possible for increased reliability of the joint, a new theory has proved the advantages of a loosely fitting thread. It allows some self-alignment of the nut relative to the threaded part of the bolt, which makes for proper functioning of the joint. In addition, increased clearances in a loosely fitting thread promote a uniform load distribution among the threads, which, in turn, improves the joint's strength.

In critical applications, extensive use is made of self-aligned fasterners. Fig. 35 illustrates some self-alignment methods (in order of increasing self-alignment). These methods are: the introduction of annular recesses in nuts and bolt heads (Fig. 35, *I*), the use of soft metal washers (Fig. 35, *II*), the use of nuts with spherical bearing surfaces (Fig. 35, *III*), the placement of spherical seat washers under nuts (Fig. 35, *IV*, *V*), etc.

The best self-alignment effect is produced when spherical seat washers are placed both under the nut and the bolt head (Fig. 35, *VI-VIII*). The radius of sphere in such washers is taken to be $R = (1.5 \text{ to } 2.5)\, d$, where d is the thread diameter.

The Tightening of Bolts. When its nut is being tightened, the bolt should be positively locked against rotation. In an assembly where its head is on the underside, the bolt should also be secured against falling out. To hold the bolt head by a spanner is unwieldy and sometimes impossible because of limited space.

Some methods for locking bolts against rotation are shown in Fig. 36. The use of countersunk head fasteners (Fig. 36, *I-III*) featuring increased friction on the conical bearing surfaces is not recommend-

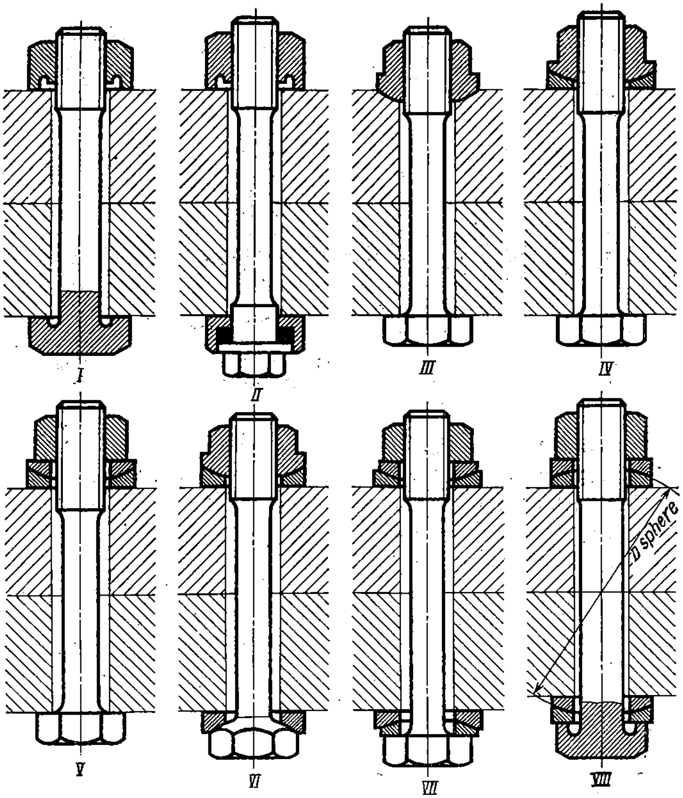


Fig. 35. Self-alignment methods used in bolt assemblies

ed for lack of positive locking. Fig. 36, *IV-X* illustrates methods for positive locking. Hexagon heads are usually fixed by placing the bolt so that one of the flats of its head is set against a suitable flat made on a component (Fig. 36, *IV*, *A*). On cylindrical (flange-like) components, circular step is provided for the same purpose (Fig. 36, *IV*, *B*). Bolts with round heads are locked with the aid of flats made on the heads (Fig. 36, *V*). Some bolt heads are provided

with a single flat (plain or recessed) extending beyond the head cylinder (Fig. 36, *VI*, *VII*). Fig. 36, *VIII*-*X* shows locking by means of a tongue made integral with the bolt head; the tongue is inserted in a seat provided in a component.

The methods shown in Fig. 36, *VI*-*X* are much more expensive than locking by flats and therefore used only in special cases.

Nibbed bolts (Fig. 36, *XI*) or square neck bolts (Fig. 36, *XII*) are now out of use for manufacturing reasons (machining the seats for square neck bolts is difficult).

Care should be taken to avoid frequent mistakes in the arrangement of locking elements. Whatever method of locking is used, eccentrical loading and weakening of the bolt head must be prevented.

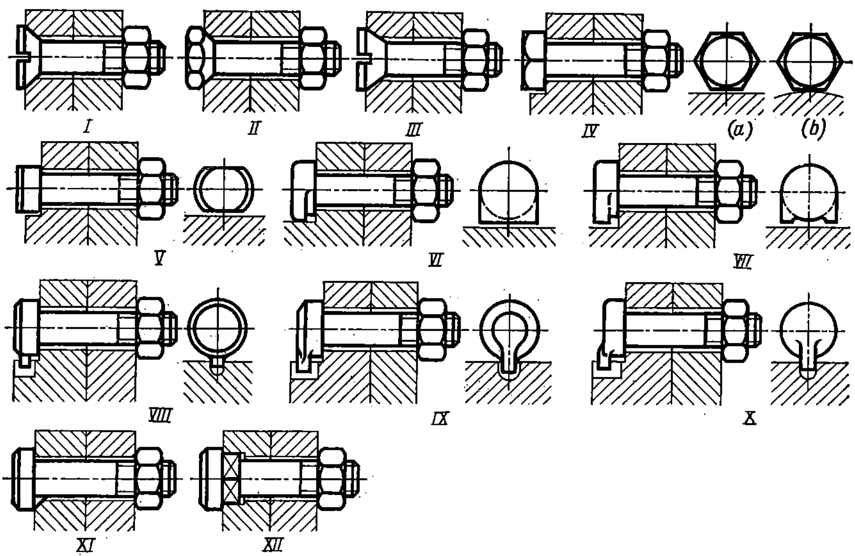


Fig. 36. Methods for locking bolts against rotation during nut tightening

Examples of improper design are shown in Fig. 37. With the bolts of Fig. 37, *I*-*III*, eccentrical loading is inevitable owing to an irregular contour of the bolt-head bearing face. In the bolt of Fig. 37, *IV*, the head is greatly weakened, and, in addition, eccentrical loading is caused by the interrupted bearing face.

Shown in Fig. 38 is a bolt whose threaded part is fixed against rotation. The shank has two slots, which engage respective teeth made in the bolt hole of a component. This method is effective for the prevention of torsional deformations of the bolt during nut tightening, which is especially important for long bolts. The method can be used only for steel components.

In addition to locking against rotation, bolts should be locked axially in the course of tightening. The axial locking is indispensable in

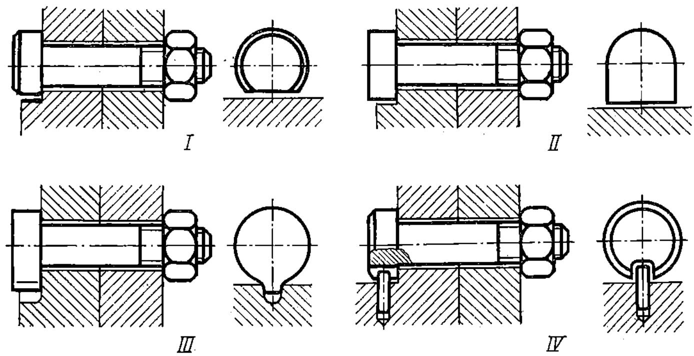


Fig. 37. Erroneous construction of elements for locking bolts against rotation

mechanized assembly operations, where nuts are tightened with nut runners. Positive axial locking of the bolt gives the best results.

Figure 39 shows some methods for axial locking of bolts (as exemplified by the fastening of a round part to a housing wall). In the design of Fig. 39, *I*, *II*, locking is effected by retaining rings placed in a circular groove on the bolt shank. In the designs shown in Fig. 39, *III*, *IV*, all the bolts of the assembly are locked by a single element, either by a retaining ring (Fig. 39, *III*) or by a disc (Fig. 39, *IV*), which also prevents the bolt from shifting rotationally.

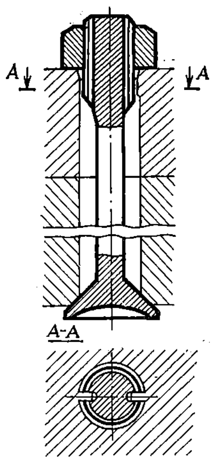


Fig. 38. Bolt locked against rotation by means of slots on its threaded portion

Figure 39, *V* shows an arrangement which provides locking against rotation and axial displacement. Here, the bolt is secured in the housing wall by a nut. A similar result can be obtained by using a stud (Fig. 39, *VI*) instead of a bolt.

The mounting of retaining rings for locking bolts axially as shown in Fig. 39, *I*, *II* deserves to be elaborated on. Fig. 40, *I-III* illustrates an erroneous mounting of retaining rings. These are disposed in recesses on a housing wall and are very hard to install on bolts inserted therein. To facilitate assembly, the recess diameter would have to be increased at least to the size of an opened retaining ring (Fig. 40, *IV*).

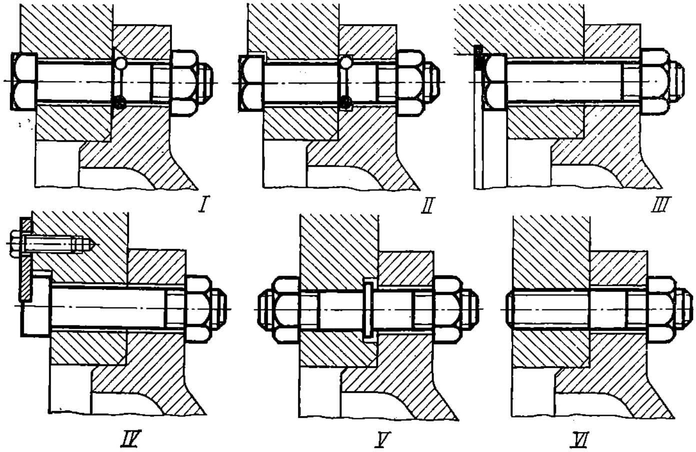


Fig. 39. Methods for locking bolts axially

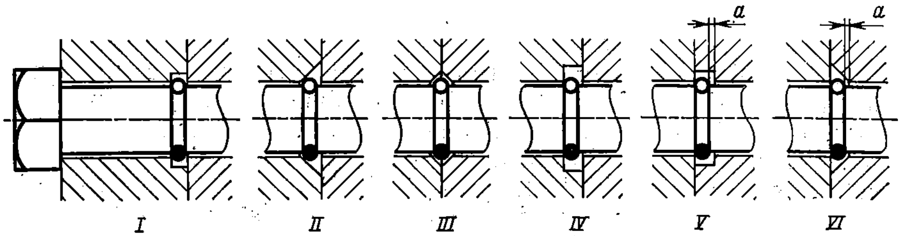


Fig 40 Methods for mounting of retaining rings for axial locking
I-III—incorrect; *IV-VI*—correct

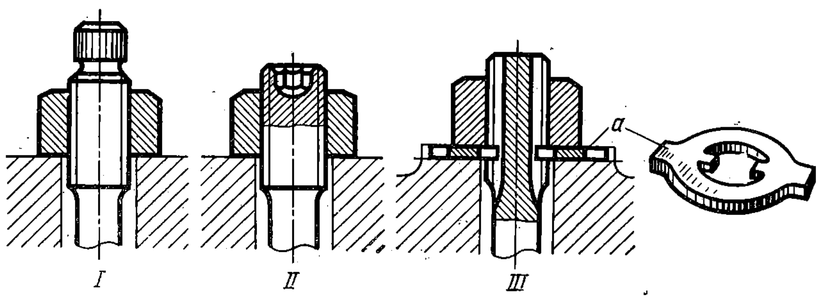


Fig. 41. Methods for prevention of bolt twisting in course of nut tightening

Correct methods of mounting retaining rings are shown in Fig. 40, *V*, *VI*. Here, recesses are made in components fastened to the housing wall, and its flat surface allows unimpeded mounting of the rings. It is important that clearance *a* between the retaining ring and the recess bottom should be provided for tight fastening of the component.

With long bolts, care should be taken to prevent twisting of the bolts by a tightening torque. To this end, special portions for spanners or keys are provided on the threaded ends of bolts (Fig. 41, *I*, *II*), or the bolt end is locked against rotation by washer *a* (Fig. 41, *III*) whose teeth engage respective slots in the bolt and in the component. Another method is shown in Fig. 38.

Tightening force is of prime importance to the strength and functioning of the bolt and nut assembly. The necessary magnitude of tightening force is found by calculation or experimentally. In critical applications, the tightening force is controlled by a torque wrench or by measuring elastic elongation of the bolt (this method is more accurate). In the latter case, the bolt shape should allow for the measurements to be taken easily. Thus, for instance, the end and head of the bolt should be provided with spherical extensions to measure the bolt length for elongation with a micrometer (Fig. 42, *I*), or conical seats to accommodate balls placed therein to effect measurements (Fig. 42, *II*).

A method of controlling the tightening force with the aid of an 'indicating' washer is illustrated in Fig. 43. Washer *a* made of a plastic material and calibrated in thickness is placed between two plain washers under a nut. Indicating washer *b* is set coaxially with the calibrated washer. The thickness of washer *a* is greater than that of washer *b* by a strictly specified amount *s* which determines, together with plastic characteristics of the material of washer *a*, the tightening force.

In the process of tightening, washer *a* gets flattened. As long as clearance *s* exists, washer *b* can be easily turned. The tightening is stopped when washer *b* can no longer be turned by hand, which signifies that clearance *s* has been taken up and the specified tightening force has been reached.

Figs. 44 and 45 show some varieties of special bolts.

Cap Screws. These are similar in construction to bolts used with nuts, although these two kinds of fasteners differ functionally.

Most of the bolt types shown in Figs. 44, 45 can be used as cap screws.

A plain hexagon cap screw is shown in Fig. 46, *I*; elastic cap screws, in Fig. 46, *II*, *III*; a cap screw assembled to a threaded insert (installed in a light-alloy component), in Fig. 46, *IV*.

The main types of threaded inserts and methods for their mounting into holes are illustrated in Fig. 47. The inserts are made of steel (and sometimes of bronze) and screwed into tapped holes with an interference-fit thread, usually by means of special drivers.

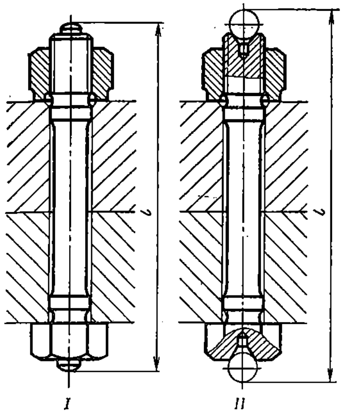


Fig. 42. Bolts adapted for measurement of their elongation due to nut tightening

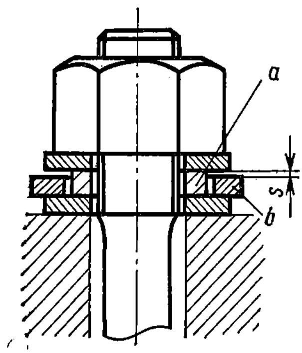


Fig. 43. Control of tightening force by means of indicating washer

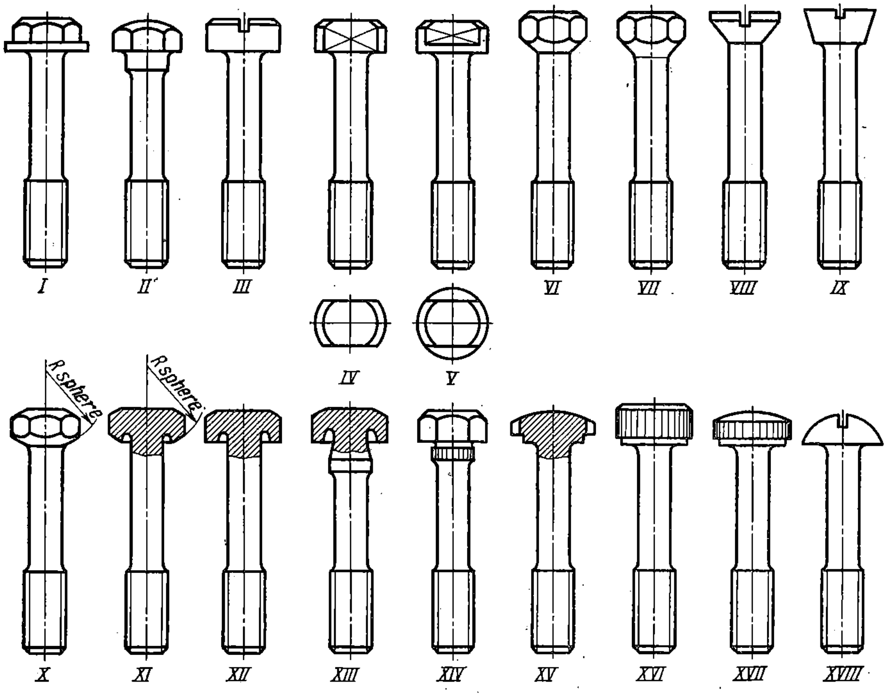


Fig. 44. Nonstandardized bolts

I—with hexagon washer head; *II*—with hexagon oval head; *III*—with slotted cheese head; *IV*—with cylindrical head and flats; *V*—with cylindrical head and flats for part of head height; *VI* and *VII*—with hexagon head and conical bearing surface; *VIII* and *IX*—with countersunk head; *X* and *XI*—with hexagon head and spherical bearing surface; *XII*—with hexagon head and recess on its underside; *XIII*—with hexagon head, underside recess and aligning collar; *XIV*—with hexagon head and knurled collar; *XV*—with splined oval head; *XVI* and *XVII*—with serrated head; *XVIII*—with slotted round head

After the insert has been set in place, its upper face is machined together with the surface of the component to ensure that the part fastened to it will seat tightly on the joint face (Fig. 47, *I*). To facilitate manufacture, threaded inserts are set with a drop-off relative to the previously machined surface of a component (Fig. 47, *II-VI*). The inserts are screwed in to bear against the bottom of a tapped hole (Fig. 47, *I*), its runout threads (Fig. 47, *II*), the insert shoulde

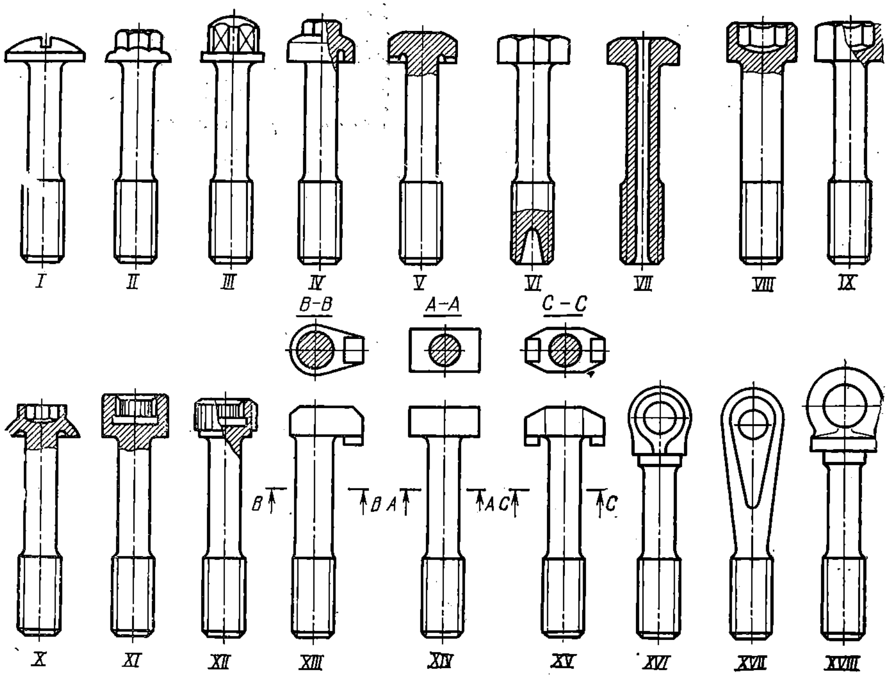


Fig 45 Special-purpose bolts

I—with slotted truss head; *II*—with hexagon spherical skirt head; *III*—with square washer head; *IV*—with reduced hexagon head and underside recess; *V*—with hexagon head and underside lock teeth; *VI*—with recess in shank end; *VII*—with through bore; *VIII* and *IX*—with hexagon socket head; *X*—with hexagon socket head and spherical skirt; *XI*—with splined socket head; *XII*—with splined socket head and external serrations; *XIII*—with hooked head; *XIV*—with T-head; *XV*—with T-head and underside tongues; *XVI*, *XVII*—swing bolts; *XVIII*—eye bolt

(Fig. 47, *III*), or against plain unthreaded portions at the insert upper end (Fig. 47, *IV*).

Fig. 47, *V* shows an insert with a neck, which allows a uniform load distribution among the threads. An insert installed from the underside of a component having a through tapped hole is shown in Fig. 47, *VI*.

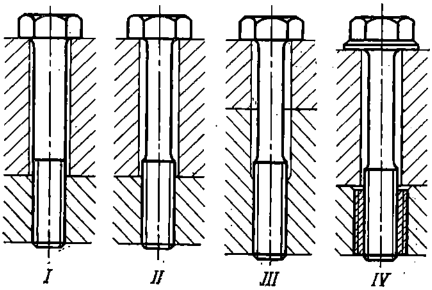


Fig. 46. Main types of hexagon cap screws

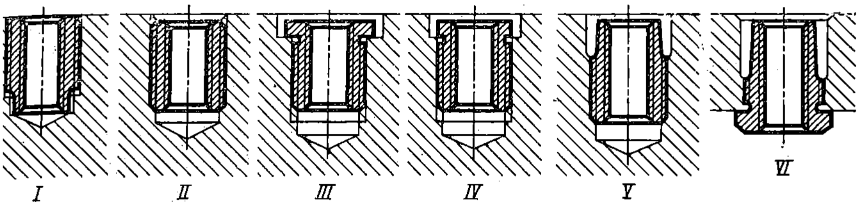


Fig. 47. Threaded inserts

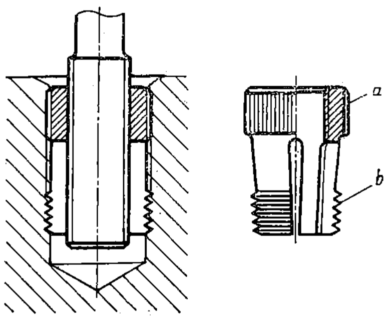


Fig. 48. Self-locking thread insert

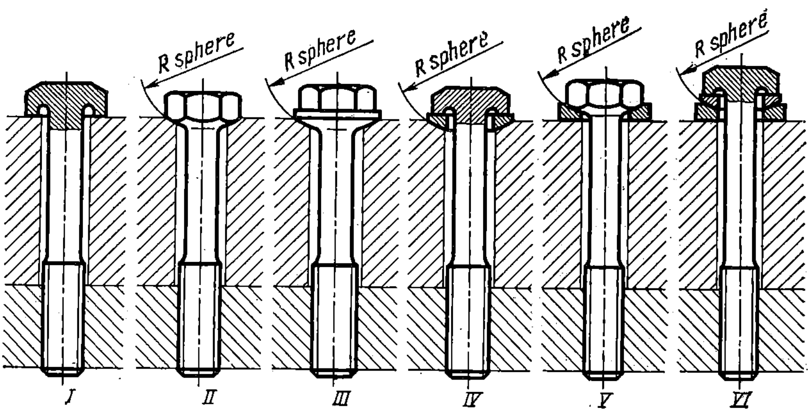


Fig. 49. Cap screws mounted for self-alignment

A threaded insert used for components made of soft metals and plastics is shown in Fig. 48. The insert has serrated collar *a*, a number of annular ridges *b* of triangular profile, and several axial slits cut on its periphery. The insert portions formed by slitting are bent inward, and then the insert is hardened. The insert is driven in a hole so that the serrations are pressed into the walls of the hole. In the course of assembly, the screw expands the split end of the insert. As this takes place, the annular ridges bite into the walls of the hole, and the insert gets locked therein.

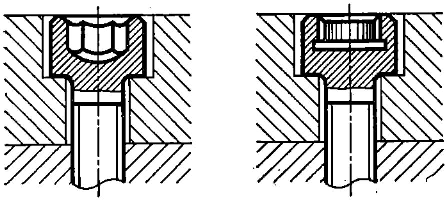


Fig. 50. Setting socket-head cap screws

With cap screws, it is desirable that their heads should be capable of self-alignment on the bearing surface. This requirement is more important to cap screws than to other types of fasteners because with bolts, the fastened components contact only the annular bearing faces of the bolt and the nut, which simplifies their self-alignment, and with slender studs, self-alignment is facilitated by their elasticity.

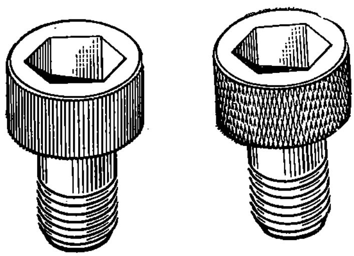


Fig. 51. Knurled soc et-head cap screws

Figure 49 shows self-aligning cap screws. The screw of Fig. 49, *I* is provided with a circular recess on the head underside, whereby a certain degree of self-alignment can be achieved. Spherical bearing faces (Fig. 49, *II-VI*) are the most effective in this respect.

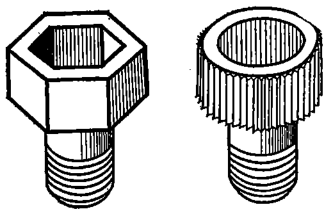


Fig. 52. Cap screws with internal and external driving elements

In applications where overall dimensions or appearance leave no place for projecting screw heads, use is made of hexagon socket head screws or socket screws with small serrations. Here, the screw is placed in a counterbored hole made in a component (Fig. 50).

In order to facilitate the initial turning of a socket screw by hand, its head is often knurled (Fig. 51).

Figure 52 shows cap screws which can be handled by both male or female head spanners.

1.6. Studs

Shown in Fig. 53 are the main types of studs. A plain stud (Fig. 53, *I*) whose shank is equal in diameter to the thread is now going out of use; only short studs are made in this form. The drawbacks of such a stud are stiffness, a disadvantageous mass-to-size ratio, difficulty in processing by such high-production methods as thread rolling, milling, and grinding (for precision screw threads), etc. Use is often made of the studs according to Fig. 53, *II*, *III* with a reduced diameter of the shank's plain portion which is either equal to or smaller than the thread minor diameter (on the average it is made equal to 0.6-0.8 the thread major diameter). The advantages of such studs are equal strength of the plain and threaded parts of the shank, elasticity, smaller mass, easy processing by high-production methods, etc.

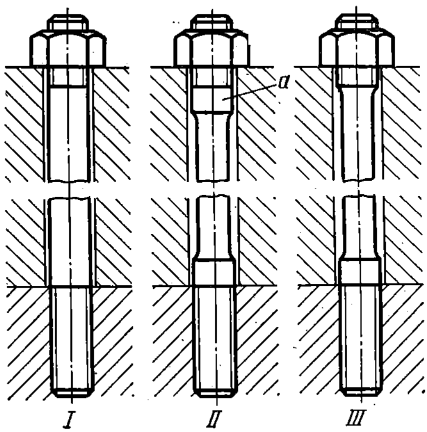


Fig. 53. Types of studs

Plain portion *a* on the nut end of the stud (Fig. 53, *II*) used in earlier stud designs is now disposed of; the threaded portion is smoothly joined to the shank's reduced part by a fillet (Fig. 53, *III*). That simplifies threading operations during stud manufacture, because the thread can now be formed by traverse machining.

The depth to which a stud can be screwed into a component, usually a housing-type part, depends on the housing material (Fig. 54, *I-IV*). Actually, in critical applications this depth is taken to be much greater than that given in Fig. 54.

Coarse threads with minimum pitches of 1.25-1.5 mm are used for studs' metal ends assembled to housings made of brittle materials (grey iron) or soft materials (aluminium, magnesium, zinc alloys, etc.). For the nut ends of studs, use may be made of fine threads (in the case of large-diameter studs).

Where both the metal and the nut ends of a stud are the same in shape and have the same size of thread (Fig. 55, *I*), errors may be made in assembly. To prevent them, the metal end is marked by various methods, e.g. its end face is rounded (Fig. 55, *II*) or indented (Fig. 55, *III*, *IV*). But the best preventive measure is to use threads of different diameter or pitch for each end of the stud.

The method of setting threaded studs has a considerable effect on the strength of stud assemblies. Three methods of setting are employed:

(1) a stud is screwed in until the thread runout engages with the top threads of the hole (Fig. 56, *I*); the method is known as 'shouldering';

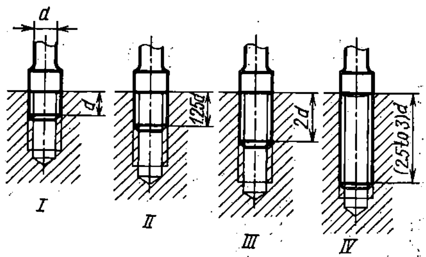


Fig. 54. Determining depth of stud driving

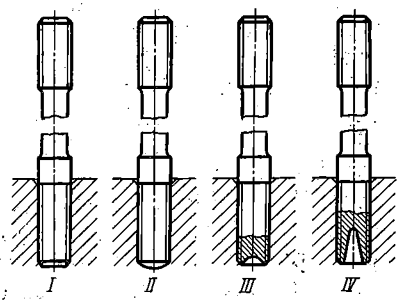


Fig. 55. Forms of stud metal ends

(2) a stud is screwed in against the bottom (Fig. 56, *II*, *III*) or runout threads (Fig. 56, *IV*) of the tapped hole; the method is known as 'bottoming';

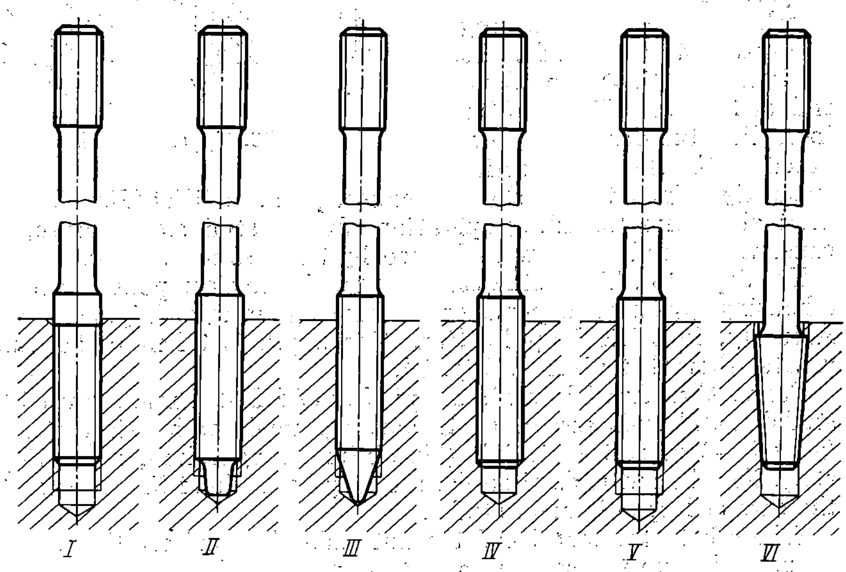


Fig. 56. Methods of stud setting

(3) a stud is set using an interference-fit thread (Fig. 56, *V*) or self-locking thread (Fig. 56, *VI*).

Setting according to the first method gives rise to tensile stresses in the stud shank, which are maximum at the upper threads and

decrease towards the lower threads. Compressive stresses develop in the housing material with a similar distribution along the thread axis. When a nut is tightened on the nut end of the stud during assembly, the stud receives additional tensile stress, and the housing additional compressive stress caused by the component being fastened to it. As a tensile working load acts on the assembly during its service, the tensile stresses in the stud further increase, whereas the compressive stresses in the housing fall, since the working load lowers the pressure of the fastened component upon the housing and produces tensile stresses therein.

In setting by the second method, compressive stresses develop in the stud shank; they are maximum at the stud end and decrease towards the upper threads. The housing material is subjected to tensile stresses, which are distributed along the thread axis in a similar way. In the course of assembly of the housing with the mating part, the tightening of the nut on the stud produces tensile stresses at the stud upper threads, whereas the compressive stresses at the stud end drop to some extent. The pressure of the fastened mating part gives rise to compressive stresses in the housing material and reduces the tensile stresses there at the bottom of the tapped hole. The action of a tensile working load on the assembly during service increases the tensile stresses at the upper threads of the stud. The compressive stresses in the housing are lowered because the working load reduces the pressure of the mating part on the housing. However, the tensile stresses rise at the hole bottom.

It can be concluded that with the first method working stresses are higher in the stud and lower in the housing than they are with the second method of stud setting. Hence, the first method is more suitable for housings of low-strength materials (e.g. aluminium and magnesium alloys), and the second, for housings of strong materials (e.g. steels). Since studs are mainly set in light-alloy housings, the first method is more common than the second.

The third method hardly develops any additional stresses in the stud and in the housing. Compressive stresses in the stud and tensile stresses in the housing caused by an interference-fit thread are insignificant with the amount of interference used. Consequently, this method is the most advantageous in respect of strength.

As distinct from the first method of setting, which provides a definite axial location of the stud, the interference-fit mounting requires that the depth to which the stud is screwed in should be checked to ensure that the stud's nut end extends a predetermined amount above the component being fastened.

The setting of studs with a taper thread (Fig. 56, *VI*) provides the same strength of joint as that of studs with an interference-fit thread, but it is used only where the axial location of the stud's nut end is not strictly specified.

Wherever possible, the metal end of the stud is additionally secured with a nut (Fig. 57, *I*) for improved strength of the assembly. Studs can also be mounted similarly to bolts with nuts (Fig. 57, *II*, *III*).

Shown in Fig. 58, *I-VIII* are studs designed for bearing against the face of the housing.

Studs are usually driven into tapped holes with the aid of stud drivers screwed on the nut end (Fig. 59). This procedure may cause the twisting of the studs and elongation of their thread, which is inadmissible for precision threads. In addition, assembly is slow because it takes time to screw the nut driver on and off the stud. A better method is to grip the stud by the flats (Fig. 58, *IV*) or hexagon (Fig. 58, *V*, *VI*) disposed at the stud's metal end. The component to be fastened, however, should have counterbored recesses to accommodate these elements, which takes additional machining operations.

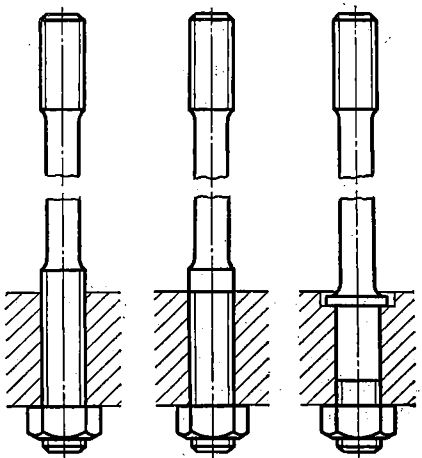


Fig. 57. Studs secured with nuts on metal end

The most easily adapted for mechanized assembly is a method whereby the stud to be driven is gripped by a plain portion which adjoins the nut end or, better still, the metal end (Fig. 58, *VIII*). Here, driving is effected by means of special stud driving devices with eccentric jaws or self-drawing rolls. For this purpose, studs should be provided with plain portions *a*, whose length should correspond to the size of the stud-driver head.

Fatigue strength of thread assemblies with studs subjected to high cyclic loads is increased by lengthening the stud's metal end (Fig. 60, *I*), by introducing relieving necks and collars where the thread joins the plain portion of the shank (Fig. 60, *II-IV*), by introducing recesses in the housing (Fig. 60, *V*), and by locating the stud deeper in the housing (Fig. 60, *VI*). The most effective method is to increase the diameter of the thread (Fig. 60, *VII*), but often it cannot be used because of dimensional limitations.

To prevent their loosening, studs are set by an interference fit; in many cases they are also locked in position.

Figure 61 illustrates some methods for locking studs in a housing. Shown in Fig. 61, *I* is a method whereby the housing material is crimped around the stud by a tubular tool. In the arrangement of

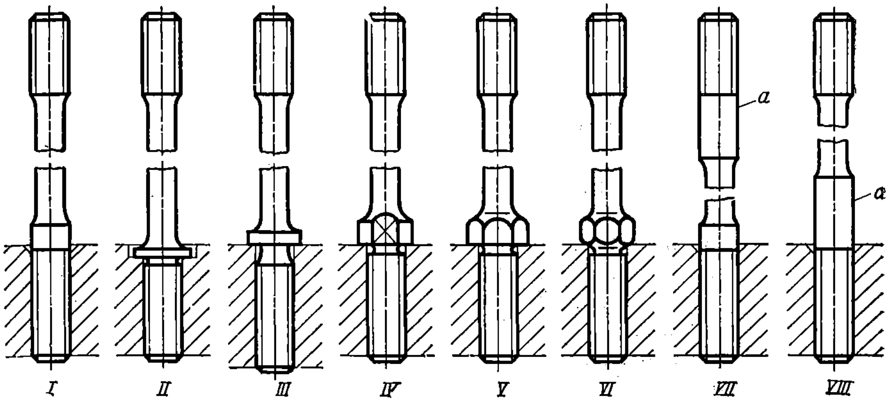


Fig. 58. Forms of shouldered studs

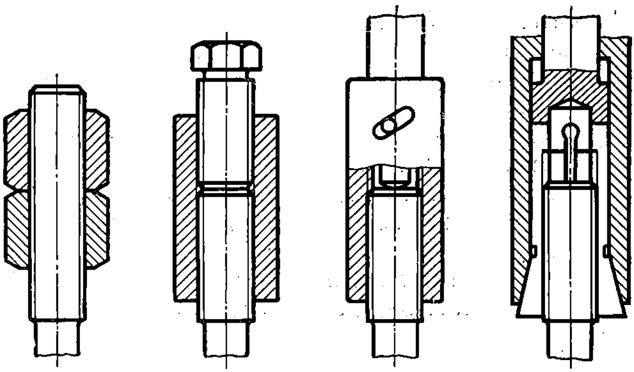


Fig. 59. Stud drivers

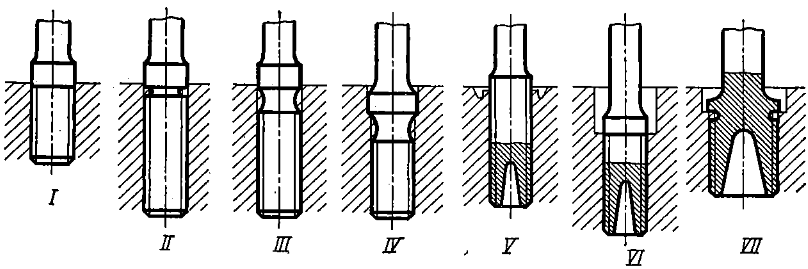


Fig. 60. Methods for increasing fatigue strength of stud assemblies subjected to high cyclic loads

Fig. 61, *II*, locking is achieved by introducing into the tapped hole an elastic (e.g. nylon) insert which seizes the stud.

In the stud shown in Fig. 61, *III* the metal end has two portions, with the lower portion being slightly upset relative to the upper portion to produce a locking effect. A self-locking stud (Fig. 61, *IV*) has a conical pin *a* inserted into a seat in its split metal end. As the stud is screwed in, the pin comes to bear against the bottom of the tapped hole and expands the split end which thereby locks the stud in the hole. The self-locking stud of Fig. 61, *V* is intended for use in components of plastic metals. The thread is separated from the shoulder by a neck; while the stud is driven in a tapped hole, the shoulder crumples the top threads of the hole into a fold which fills the circular space formed by the neck and so locks the stud. The similar effect is produced by a stud with a tapered shoulder.

In soft-metal components, stud shouldering gives rise to plastic deformations of the material in the form of a lip on the surface around the stud (Fig. 62, *I*). To eliminate this effect and provide for tight contact between the fastened components, the tapped hole is chamfered (Fig. 62, *II*) or recessed (Fig. 62, *III*). Sometimes, chamfers are provided in both the housing and the component fastened to it (Fig. 62, *IV*).

When studs, especially those with an interference-fit thread, are driven into blind tapped holes, they compress the air within the closed space underneath. That may prove to be dangerous, especially in view of the fact that the specific volume of compressed air sharply increases with rising temperature. Bosses with blind tapped holes have been known to burst occasionally under the pressure of the compressed air inside.

In order to avoid that, such bosses are provided with ducts for air escape (Fig. 63, *I*, *II*). In some cases, the air comes out through grooves (Fig. 63, *III*) or bores (Fig. 63, *IV*) made in studs (only in short ones). The last two methods (Fig. 63, *III*, *IV*) are not recommended since they weaken the studs.

Sometimes, the solution is to increase the volume of the closed space between the stud end and the tapped hole bottom by deepening the hole and recessing the stud end (Fig. 63, *V*). The volume of this space is determined with regard to the laws of thermodynamics so as to avoid critical air pressures when driving the stud.

Studs are set in soft-metal housings with the aid of threaded inserts (Fig. 64, *I* and *II*); the latter are made of steel (or sometimes of bronze) and screwed into the tapped holes, as a rule with an interference fit. Fig. 64, *III*, *IV* shows threaded inserts with elastic necks for uniform load distribution among the stud threads. Fig. 64, *V* illustrates the locking of a threaded insert in a tapped hole. The split end of the insert is radially expanded by the tapered point of the stud end, which acts on a retaining ring set into the insert internal thread.

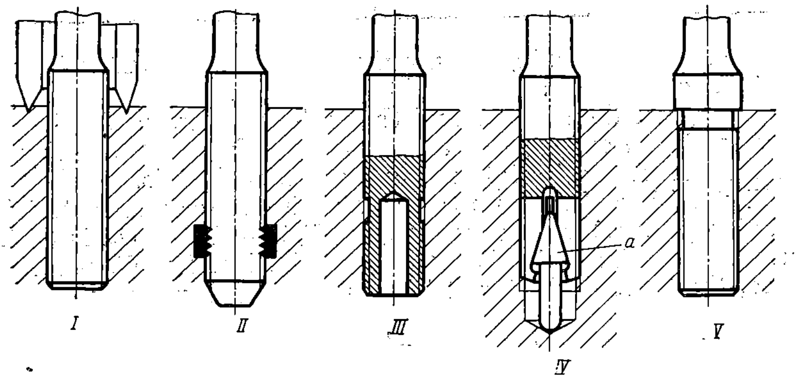


Fig. 61. Methods for stud locking

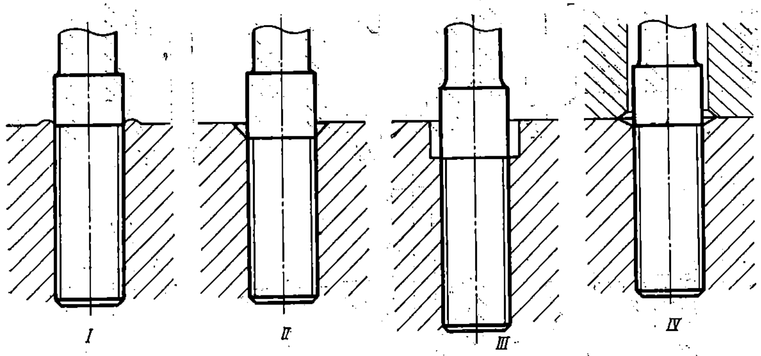


Fig. 62. Methods for preventing formation of lip around stud after its driving

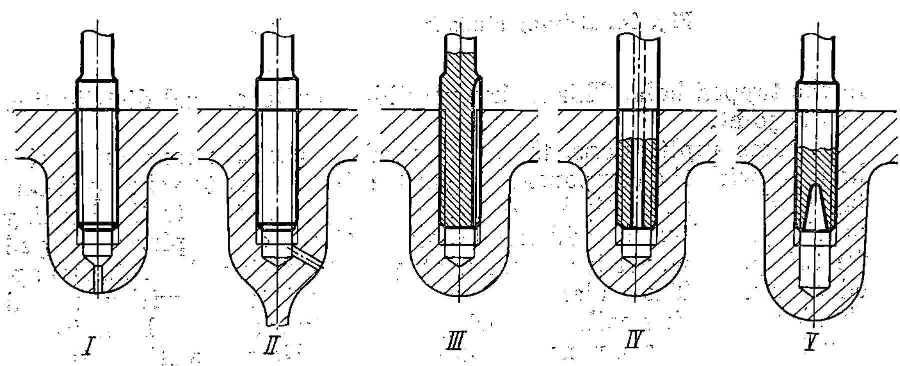


Fig. 63. Preventing air compression in deep stud holes

A method for locking at once the insert and the stud is illustrated in Fig. 64, *VI*. The segments of the insert formed by slitting are bent inwards, and then an external thread is cut thereon. As the stud is screwed in, its end expands the segments of the insert, and that creates interference in its internal and external threads.

Figure 64, *VII* shows a self-locking insert for soft materials (including plastics). In the assembly of Fig. 64, *VIII*, use is made of a screw thread insert in the form of a helical coil of diamond-shaped wire; the turns of the insert fill the thread grooves on the stud and

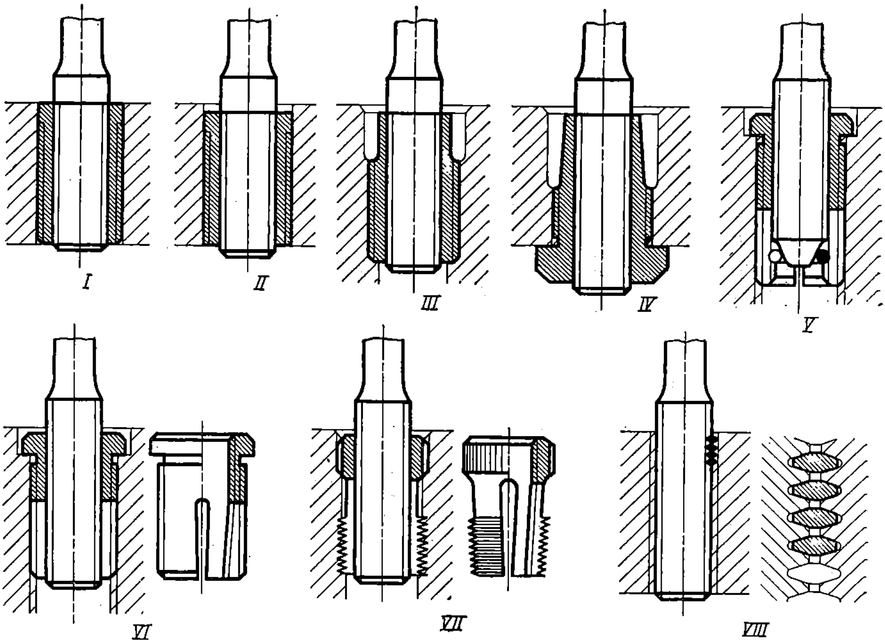


Fig. 64. Driving studs into threaded inserts

in the tapped hole. This makes possible a uniform load distribution in the joint.

In some applications, there is a need for positive connection of a housing with the component fastened to it in order to take up shearing loads or to strictly locate the component relative to the housing on the joint face. In addition to conventional dowel pins employed for this purpose, use is also made of special locating elements arranged on studs. These elements may be made integral with the stud shank as centring collars entering accurately machined bores in the housing and the component (Fig. 65, *I*, *II*). Here, a difficult task of screwing the stud and simultaneously centring its collar in the bore

is usually accomplished by using a loose thread for the stud's metal end.

A better design is a locating element in the form of a bushing mounted coaxially with the stud (Fig. 65, *III*, *IV*).

Figure 65, *V*, *VI* presents instances wherein two components fastened to a housing are located relative to each other and, at the same time, to the housing.

Like other fasteners, studs are initially loaded when the nuts are tightened on them during assembly, and the amount of the initial

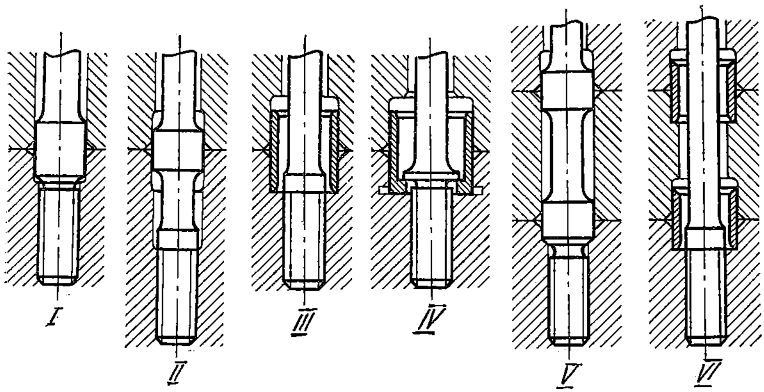


Fig 65 Studs with locating elements

loading has an effect on the strength, tightness and functioning of the assembly. The amount of preloading is determined by calculation or experiment; it depends on the material of parts to be fastened, the relation between elasticity of the stud and that of the fastened parts, operating conditions, the required tightness of the joint, and, finally, the working temperature.

In critical applications, the amount of preloading is strictly checked. Tightening is carried out with torque wrenches. The sequence of nut tightening on individual studs is specified for multi-stud assemblies; the nuts are tightened in two stages (preliminarily and finally), with the prescribed tightening procedure being strictly observed.

When tightening nuts on slender elastic studs, the latter are subject to twisting by the moment of friction forces in the thread. That, on the one hand, gives rise to undesirable, sometimes very high, stresses in the stud, and, on the other, makes it difficult to ensure the specified tightening force. In this case, the torque wrench will register the force of stud twisting rather than tightening.

Where nuts torqued on studs are locked 'on the component', account must be taken of another factor: the stud twisted in the course

of nut tightening gradually recoils due to vibrations, pulsating loads, etc., and, turning within the nut, alters the initial tightening force.

With long elastic studs, various means are used to prevent stud twisting. For example, the nut end of a stud is provided with a slot, square, hexagon, etc., to hold the stud during nut tightening (Fig. 66, *I-IV*). This makes the assembly procedure more complicated. Another method of assembly, where the nut end of the stud is permanently locked against twisting by washer *a* (Fig. 67), which, in turn, is locked on the housing, is more advantageous, although it requires a more complex design.

The nut ends of long studs installed in a housing often deviate from the true position (sometimes to an extent which makes it impossible to mount over them the component to be fastened to the housing). In such cases, fitters resort to straightening the studs individually for proper alignment, but this method cannot be recommended since it gives rise to additional stresses in the studs.

Several methods are used to solve this problem: the first is to hold the axes of tapped holes in the housing and of the clearing holes in the mating part strictly square to the joint face; to hold strict straightness of studs and parallelism of the pitch-diameter generator to the stud axis; and the second is to increase the elasticity of studs and to use a loosely fitting thread on both the metal end and the nut end of the stud (with subsequently locking the thread by some method).

Attempts have been made to centralize the nut ends of studs in accurately machined holes of the fastened part with centring collars on the studs (Fig. 68, *I*), and with nuts which are centred in the respective counterbores directly (Fig. 68, *II*) or through spacer bushings (Fig. 68, *III*).

However, these methods do not obviate the need for geometrical accuracy of the stud axis; on the contrary, they require that the sources of axis deviations should be eliminated. The virtue of these methods is that the studs get in place automatically, without any alignment by the fitter, as the part to be fastened is mounted or the nuts are tightened. If elastic deformations of the studs are insignificant, these methods can be regarded as acceptable, for they simplify assembly.

Figure 69 shows an arrangement where the nut end of a stud is centralized and sealed. Where studs are installed in large-diameter clearing holes (Fig. 70, *I*), measures should be taken to prevent the stud ends from 'swaying' during nut tightening. Some methods of prevention are shown in Fig. 70, *II-IV*.

Like any other types of fasteners, heavy-duty studs require the use of nuts with spherical bearing surfaces (Fig. 71, *I-IV*), which ensure self-alignment of the nuts and reduce bending of the studs.

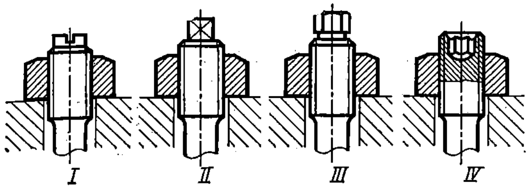


Fig. 66. Studs with features provided to prevent twisting during nut tightening

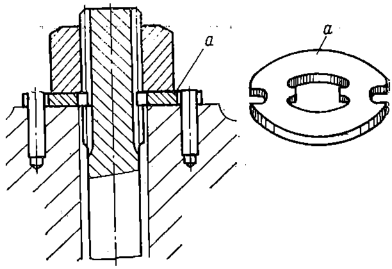


Fig. 67. Method for preventing stud from twisting during nut tightening

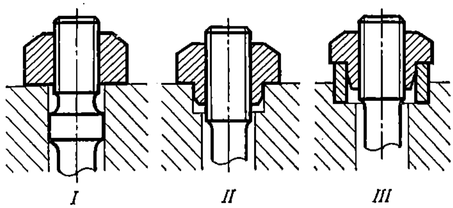


Fig. 68. Centring the nut end of stud in hole

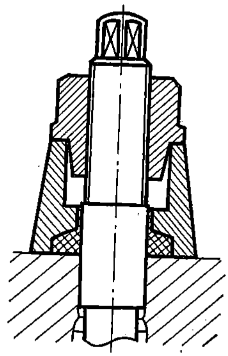


Fig. 69. Centring and sealing the nut end of stud

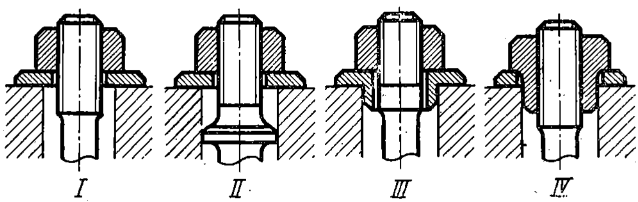


Fig. 70. Centring the nut end of stud in large-diameter holes

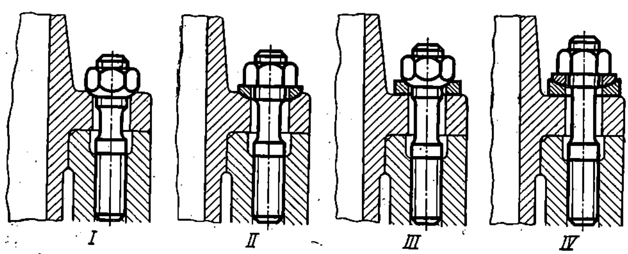


Fig. 71. Setting nuts with spherical bearing faces

1.7. Machine Screws

Slotted head machine screws are used only in assemblies which are not subjected to external loads (e.g., in instruments, and for fastening small components, brackets, clips, yokes, strips, panels, etc.). The main drawback to these fasteners is that they are impossible to tight hard and are difficult to lock.

Figure 72 shows the main types of slotted head machine screws: cheese head screws, fillister head screws (*I*, *II*); round head screws (*III*); countersunk head screws (*IV*, *V*), and raised countersunk

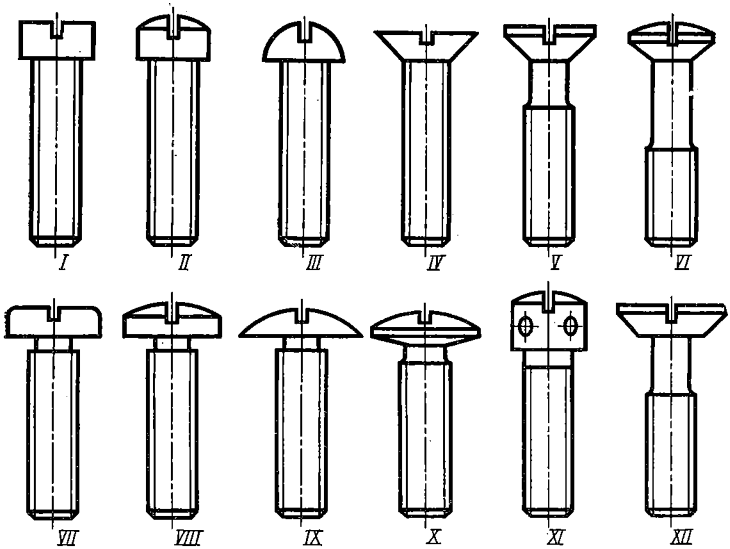


Fig. 72. Main types of slotted head machine screws

head screws (*VI*). Some modifications of these screws are shown in Fig. 72, *VII-XII*.

Of all these varieties, the most attractive for the machine builder are countersunk and raised countersunk head screws, which enable him to design assemblies without jutting screw heads. Unfortunately, countersunk head screws and their modifications have much more disadvantages than the other types. The main drawback is the difficulty of simultaneous location on two different surfaces, i.e. on the thread and on the conical head. It is especially so in assemblies with several screws. As a rule, the coordinates of tapped holes in a component and those of respective clearing holes in its mating part have deviations due to inevitable machining errors; consequently, only one of the screws in the assembly sits in its countersink correctly, whereas the heads of the other screws are misaligned in their coun

tersinks. This, however, can be alleviated by using a loose screw thread. Another drawback is the difficulty of locking. While cheese head or round head screws are relatively easy to lock with spring washers (or with a wire binding, as for the screws shown in Fig. 72, *XI*), the problem of locking countersunk head screws has yet to

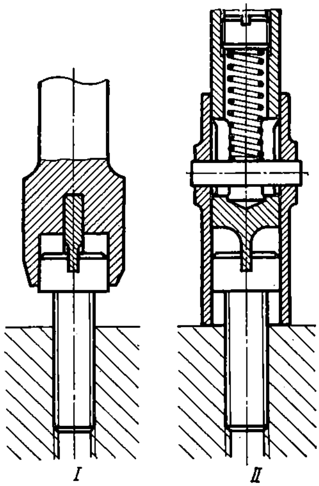


Fig. 73. Screwdrivers with centring sockets for slotted cheese head screws

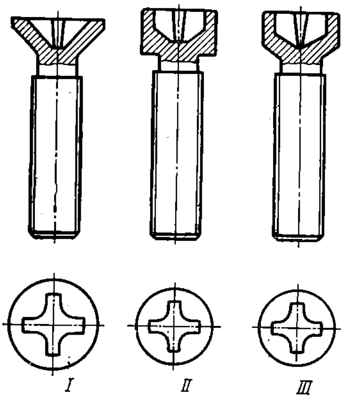


Fig. 74. Screws with cross recesses for special screwdrivers

be solved. The most reliable methods of locking countersunk head screws (staking the part material around the screw head) can only be used for parts of plastic metals, and, in addition, these methods make such joints inseparable.

The third drawback of countersunk head screws is that the slot on the screw head is a weak element (this is equally true for round head screws and, to a lesser degree, for cheese head screws). In repeated use, screwdrivers deform the slot and render the screw unfit for service.

Machine screws used in engineering are heat treated to a hardness of Rc 40-45.

Yet another drawback is difficulty in using mechanized screwdrivers, since the shape of the countersunk head and the slot does not allow proper alignment of the screwdriver tip. With cheese head screws, the alignment is relatively easy to achieve by introducing centring elements which are located on the head periphery (Fig. 73).

A step forward in the design of countersunk-head, cheese-head, and round-head screws is the introduction of heads with cross recesses (Fig. 74); such screws are driven with special-tip screwdrivers (Fig. 75, *II*).

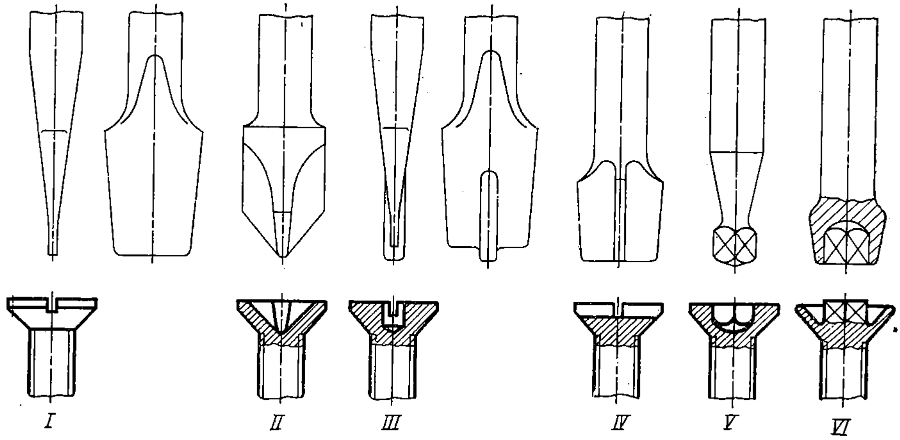


Fig. 75. Screw heads

I—slotted; *II*—with cross recess; *III*—slotted with centring hole; *IV*—with cross slots; *V*—with square socket; *VI*—with external square

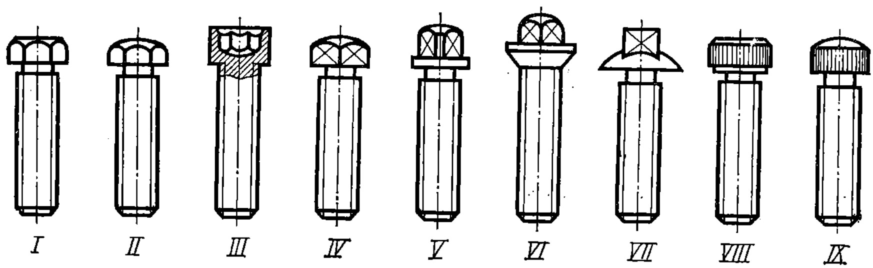


Fig. 76. Machine screws with heads designed for high torque

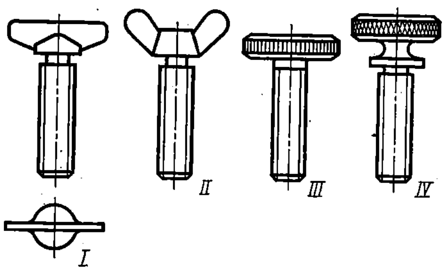


Fig. 77. Screws for hand tightening

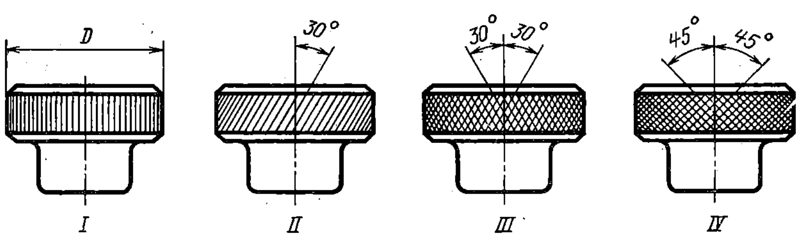


Fig. 78. Types of knurls

The screws of this type can be tightened with great pull, since the shape of the recess resists damage and simplifies both manual and mechanized driving.

The same effect is provided by other means, e.g. a slot and a centring hole (Fig. 75, *III*), a cross slot (Fig. 75, *IV*), a square socket (Fig. 75, *V*), an external square, etc.

Fig. 76 shows machine screws with heads which allow increased driving pulls to be applied for tightening: hexagon heads (Fig. 76, *I*, *II*), hexagon socket heads (Fig. 76, *III*), square heads (Fig. 76, *IV*-*VII*), serrated heads (Fig. 76, *VIII*, *IX*), etc. These screws might as well be classed as small cap screws.

The screws shown in Fig. 77 are intended for hand tightening. These are winged screws (Fig. 77, *I*, *II*) and screws with knurled heads (Fig. 77, *III*, *IV*).

Fig. 78 shows the main types of knurls: straight-line (Fig. 78, *I*), diagonal (Fig. 78, *II*) and diamond (Fig. 78, *III*, *IV*). The knurl pitch t depends on the diameter of the screw head and its material. For straight-line knurls the recommended pitches are $t = 0.5$; 0.6; 0.8; 1.0; 1.2 mm, and for diagonal and diamond knurls $t = 0.6$; 0.8; 1.0; 1.6 mm. The larger the screw-head diameter, the greater the knurl pitch.

1.8. Nuts

The main types of hexagon nuts are shown in Fig. 79. These are: full-bearing nuts with chamfers of $D_1 = S$ (Fig. 79, *I*) and $D_1 = 0.95S$ (Fig. 79, *II*); a double-chamfered nut (Fig. 79, *III*); and washer-faced nuts (Fig. 79, *IV*, *V*).

Figs. 80 and 81 show nuts of various types: a hexagon slotted nut (Fig. 80, *I*), a hexagon castle nut (Fig. 80, *II*) and its variations (Fig. 80 *III*, *IV*); a nut with a hexagon of reduced height and skirt (Fig. 81, *I*); a hexagon nut with a guiding taper for spanner (Fig. 81, *II*); hexagon nuts with conical (Fig. 81, *III*) and spherical (Fig. 81, *IV*) bearing surfaces.

Depending on applications, nuts can be from $0.3d$ to $1.25d$ in thickness (where d is the thread diameter). Nuts of smaller thickness are used as lock nuts and for small-load applications, those of greater thickness for heavy-duty applications and frequently disassembled screw joints. For medium operating conditions, use is made of nuts (0.8-1) d in thickness. This thickness-to-diameter relation approximately corresponds to the equal strength of the nut and of the threaded shank.

Shown in Figs. 82-89 are nuts of various shapes; in Fig. 90, socket-type nuts (with hexagon and serrated sockets) used where tightening has to be done with a great pull and radial dimensions are

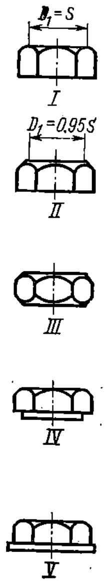


Fig. 79. Types of hexagon nuts

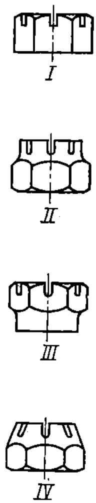


Fig. 80. Types of hexagon slotted nuts

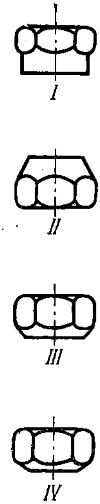


Fig. 81. Special hexagon nuts

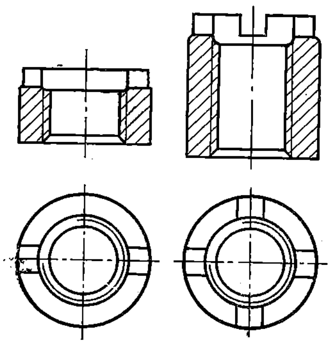


Fig. 82. Nuts with end faces slotted for spanner

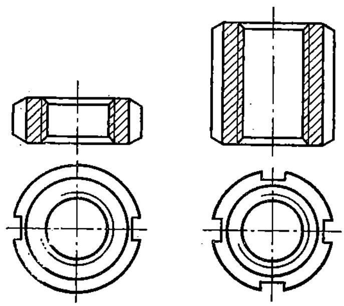


Fig. 83. Nuts with slots on circumference

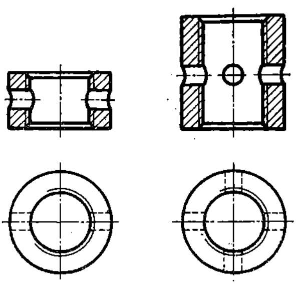


Fig. 84. Two- and four-pin driven nuts

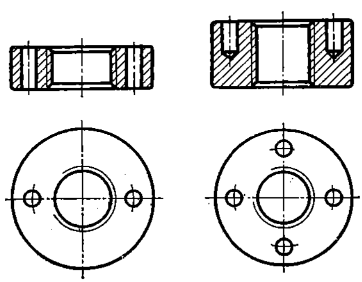


Fig. 85. Circular nuts with axial holes for face spanner

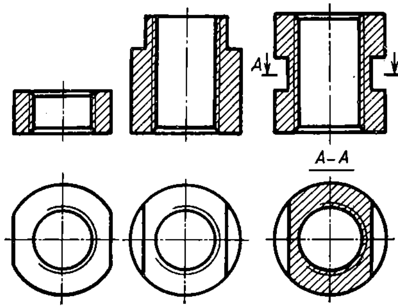


Fig. 86. Circular nuts with flats

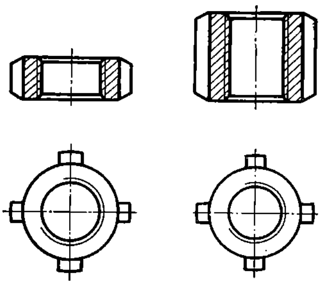


Fig. 87. Nuts with projections for spanner

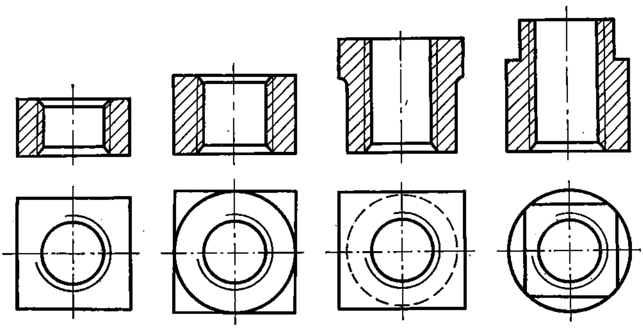


Fig. 88. Square nuts

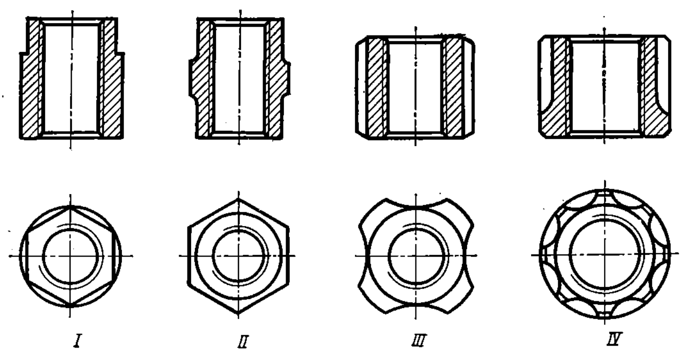


Fig. 89. Special nuts

I—with hexagon on the end; *II*—with hexagon in the middle; *III*, *IV*—milled nuts

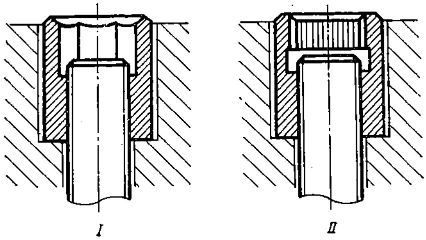


Fig. 90. Nuts with hexagon socket (*I*) and serrated socket (*II*)

limited; in Fig. 91, cap nuts used where the screw joint has to be sealed; and in Figs. 92, 93, male threaded nuts.

Serrated Nuts. Cylindrical nuts with small serrations on the periphery (Fig. 94) are promising, and in future they are likely to supersede hexagon nuts. Their principal feature is an advantageous distribution of forces in the process of tightening. As seen from Fig. 95, the arm of the tightening force applied to a 60° serration of the nut is about twice as long as that of the same force applied to the equivalent hexagon nut.

The number of serrations on the nut periphery may be 6 to 7 times that of the hexagon flats. Hence, with the same tightening tor-

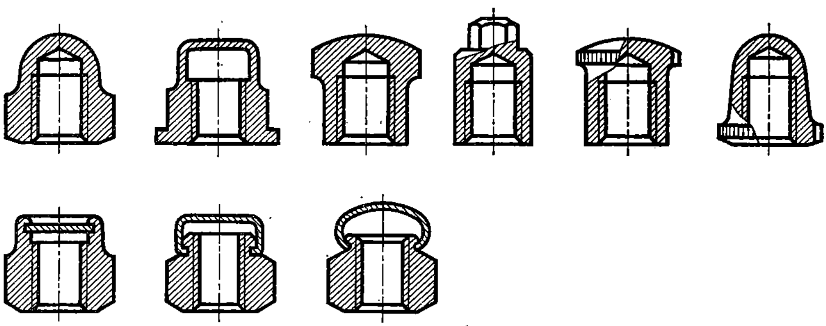


Fig. 91. Cap nuts

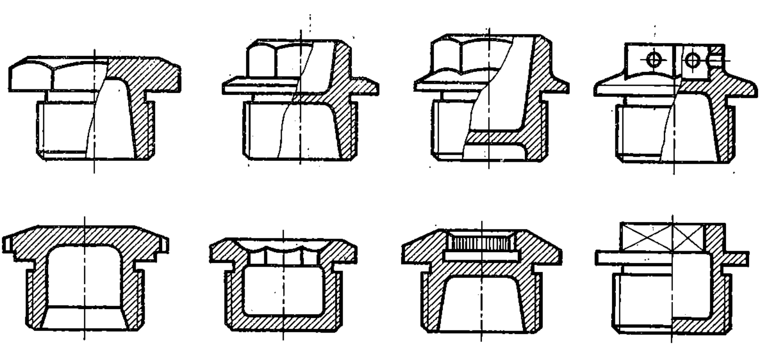


Fig. 92. Male threaded blind nuts

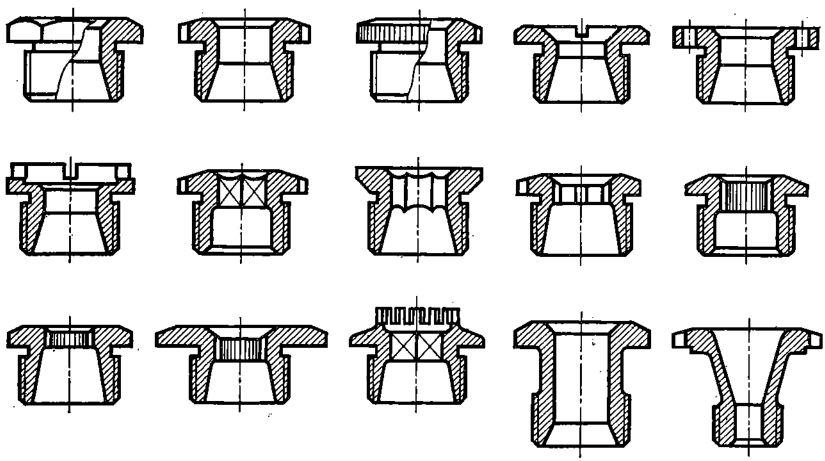


Fig. 93. Male threaded hollow nuts

que, the force for one serration will be lower than that for one hexagon flat by a factor of 12 to 15 when tightening with a closed end spanner, and by a factor of 36 to 45 when tightening with an open-ended spanner. That eliminates the danger of deforming the surfaces engaged by the spanner, which is quite real with hexagon nuts. What is more, the serrations will not let the spanner break loose from the nut during tightening.

Another advantage of serrated nuts is the possibility of turning the nut through any angle when tightening, which simplifies assembly at the places where the room for wrench clearance is small and the wrench swing is limited.

With the same thread diameter, serrated nuts have smaller radial dimensions and mass than hexagon nuts. The drawback of serrated nuts is that they can be handled only by a closed end spanner.

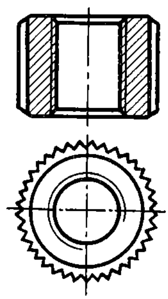


Fig. 94. Serrated circular nut

When designing assemblies with serrated nuts, a space must be provided above the nut for putting a closed-end spanner on it. The space can be reduced in height by using box spanners of smaller thickness. A reduced height of the nut serrations (Fig. 96, *I-III*) simplifies manipulations with the spanner, for it can be guided by the nut cylindrical neck when being put on the nut. Special adjustable spanners may be used which allow access to the nut from the side.

The strength of serrations (Fig. 97, *I*) features so high a safety factor that their number can be reduced without much detriment to the nut strength (Fig. 97, *II-IV*). The mass of the nut becomes lower, but the advantages of turning the nut through any desired angle are retained if the spanner socket has the mating serrations along the entire periphery.

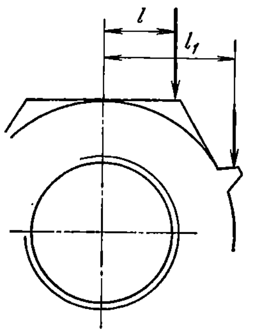


Fig. 95. Action of tightening forces in hexagon and serrated nuts

When designing serrated nuts, the following dimensional relationships are recommended (Fig. 98):

(1) the minor diameter of serrations $D_1 = (1.35\text{-}1.5)\,d$, where d is the nominal diameter of thread; the upper limit (1.5) is for small nuts, and the lower limit is for medium and large nuts;

(2) the major diameter of serrations (outer diameter of nut) $D = (1.1\text{-}1.15)\,D_1$; the upper limit is also for small nuts, and the lower limit, for medium and large nuts;

(3) the height of the nut $H = (0.8\text{-}1)\,d$ for medium loads.

Serrated nuts (Fig. 96) are most frequently locked with split cotter pins.

Circular Nuts. These are used to clamp antifriction bearings and similar components on large-diameter shafts.

The distinctive feature of circular nuts is a small height-to-diameter ratio. A nut of normal height would prove excessively strong and very heavy due to the large-diameter thread.

The height of the nut necessary to provide equal strength of the nut and shaft (for the case of a hollow shaft) can be readily calculated.

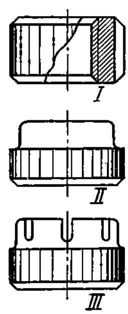


Fig. 96. Forms of serrated nuts

The equal strength of a hollow shaft loaded in tension and its threaded portion loaded in shear by the tightening force can be described by the following elementary expression:

$$\tau \pi D_p H = \sigma \frac{\pi}{4} (D_p^2 - D_h^2)$$

where τ = shear stress in thread; σ = tensile stress in shaft; H = length of loaded portion of thread (height of nut); D_p and D_h = pitch diameter of thread and diameter of hole in shaft, respectively.

Hence

$$H = \frac{\sigma}{\tau} \frac{D_p}{4} \left[1 - \left(\frac{D_h}{D_p} \right)^2 \right]$$

For medium loads, it can be assumed, taking into account stress concentrations in the thread, that the allowable shear stress in the thread is lower in magnitude by a factor of 2 than the allowable ten-

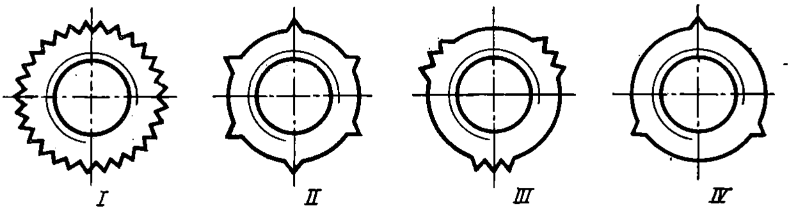


Fig. 97. Disposition of serrations on nuts

sile stress in the shaft. Then

$$H = \frac{D_p}{2}\left[1 - \left(\frac{D_h}{D_p}\right)^2\right]$$

It follows from the expression that the nut height decreases with an increase in the shaft hole diameter (Fig. 99).

When standardizing circular nuts, it is difficult to allow for ratio D_h/D_p; the nut height is usually established depending only on the diameter of thread, D. The nut height H (Fig. 100) is taken to be approximately (0.15-0.25) D, lower values being used for large-diameter nuts, and greater values for small-diameter nuts.

Because they are small in height, circular nuts are provided with fine threads only. The use of coarse threads (Fig. 101, *I*) would result in fewer thread turns on the nut (due to a smaller number of full-form threads relative to their total number), hinder the axial guidance of the nut on the shaft, and also weaken the shaft owing to a smaller minor diameter of the thread.

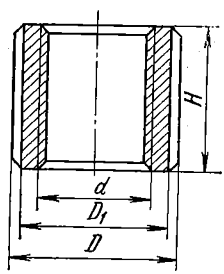


Fig. 98. Determining dimensions of serrated nut

Thread pitch s for circular nuts is usually taken to be (0.015-0.05) D, where D is the thread diameter; here, the upper limit is applied to thread of small diameter (30 to 50 mm), and the lower limit to large-diameter threads (100 to 120 mm). When designing a circular nut, its thread pitch and height should be selected so as to ensure a total number of threads of no less than 5-6 (Fig. 101, *II*).

As in all other screw joints, the thread on the shaft should be longer than that in the nut to ensure some extra thread at both sides of the nut in its nominal position; the amounts of extra thread are given in Fig. 102.

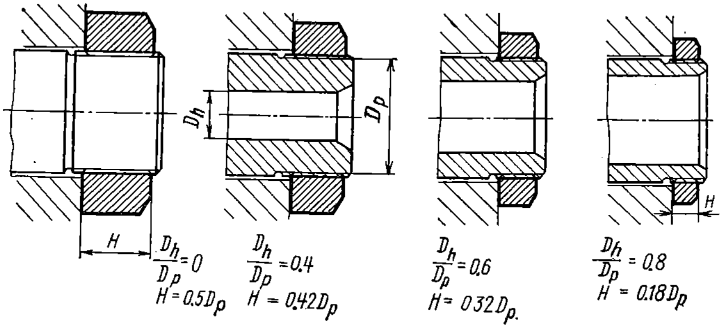


Fig. 99. Determining height of circular nuts

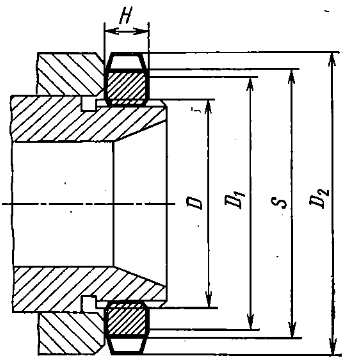


Fig. 100. Determining dimensions of circular nuts

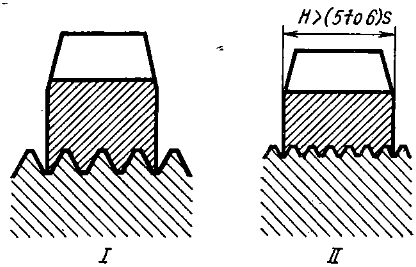


Fig. 101. Determining thread pitch for circular nuts

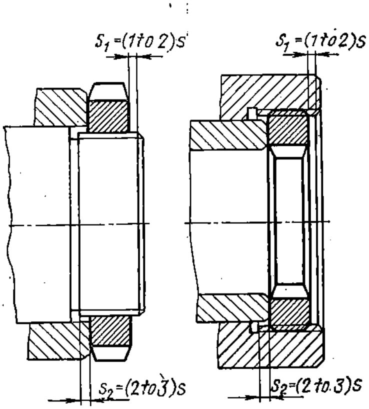


Fig. 102. Extra thread in circular-nut assemblies (s is thread pitch)

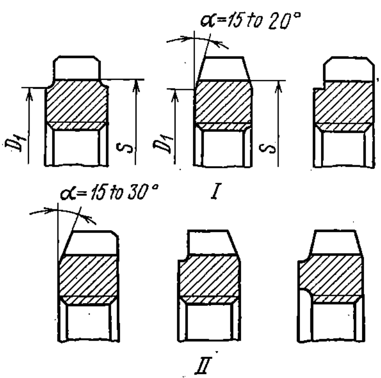


Fig. 103. Arrangement of slots relative to bearing faces in circular nuts

Dimension S, which is the distance between the bottoms of diametrically opposed slots on the nut circumference, determines the minimum thickness of the nut body; it is made equal to (1.2-1.3) D. The nut outside diameter D_2 varies within the limits of about (1.4-1.5) D (Fig. 100).

The slots on the nut should be arranged beyond its bearing faces so that burrs on the slot edges caused by spanners do not prevent the proper contact of the nut bearing surface with the clamped component. This is achieved by introducing recesses or chamfers on one or, better, both sides of the nut (Fig. 103). The outer diameter D_1 of the bearing face should be smaller than distance S between the slots' bottoms by at least 0.5-1 mm.

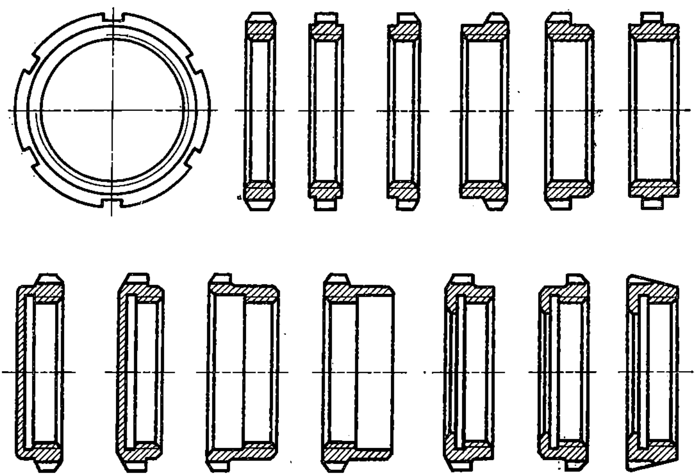


Fig. 104. Internally threaded circular nuts

Figure 104 shows internally threaded circular nuts with various arrangements of slots; Figs. 105-113 show circular nuts with driving elements of different types.

The most frequently used are nuts with external slots, whose number varies from 4 to 12. Such nuts are driven with hook spanners (Fig. 114, *I*), face spanners (Fig. 114, *II*) or closed end spanners with internal radial teeth (Fig. 114, *III*).

The number and configuration of slots on the nut circumference have a marked effect on the nut weight. In applications where the reduction of mass is essential and where a great number of circular nuts are used, special attention is paid to the design of the slots.

Figure 115 gives values of the relative mass of differently shaped nuts with reference to a nut with four slots whose mass is taken as

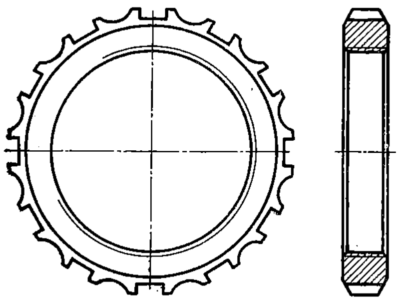


Fig. 105. Special-shape circular nut

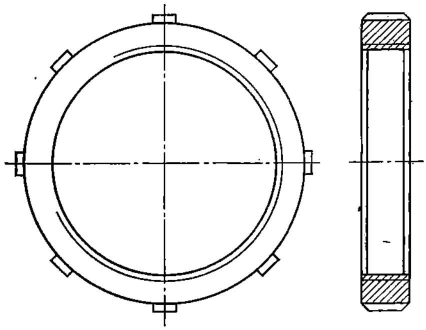


Fig. 106. Circular nut with splines for hook spanner

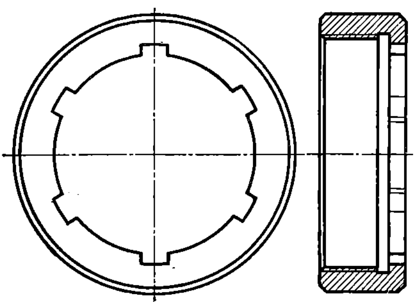


Fig. 107. Circular nut with internal slots

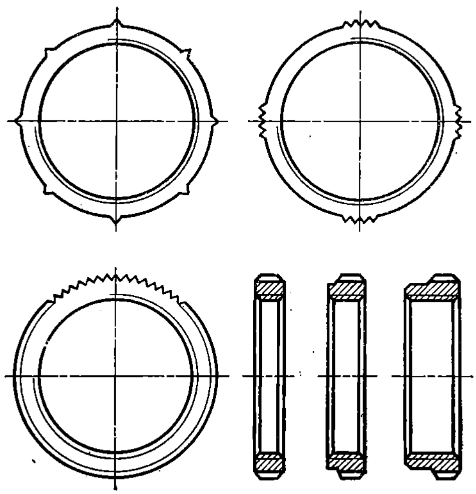


Fig. 108. Circular nuts with triangular splines

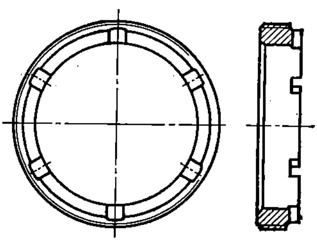


Fig. 109. Circular nut with lateral slots

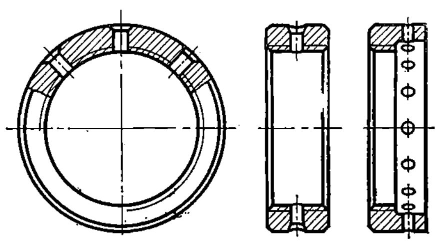


Fig. 110. Circular nuts with radial through holes for pin spanner

unity. As seen from Fig. 115, *I-IV*, the mass of the nuts can be substantially reduced by merely increasing the number of slots. The mass of the nut with twelve slots (Fig. 115, *IV*) is 86% that of the four-slot nut (Fig. 115, *I*). Further reduction of the mass is achieved by making recesses on the periphery of projections between the slots

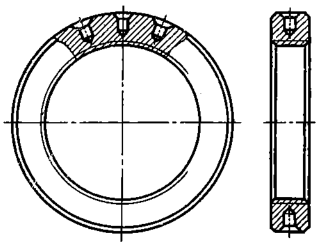


Fig. 111. Circular nut with radial blind holes for pin spanner

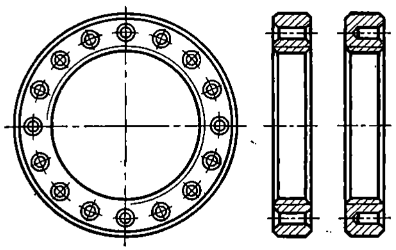


Fig. 112. Circular nut with lateral holes for face spanner

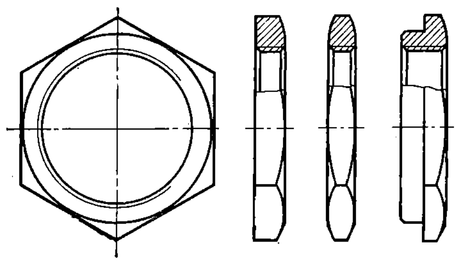


Fig. 113. Hexagonal nuts

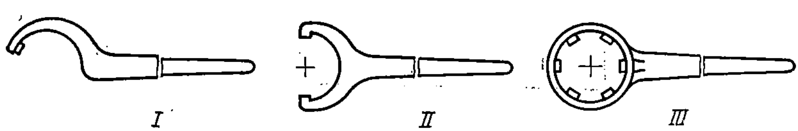


Fig. 114. Spanners for driving circular nuts with external slots

(Fig. 115, *V*), by reducing the height and width of the projections (Fig. 115, *VI*), and by reducing their number (Fig. 115, *VIII*).

The most advantageous design is shown in Fig. 115, *IX*; the nut has a small number of triangular projections, and its mass comes to 53% of the reference nut mass.

The slot configurations shown in Fig. 115, *V-IX*, can be obtained by a high-production method of generating with a form hobbing cutter.

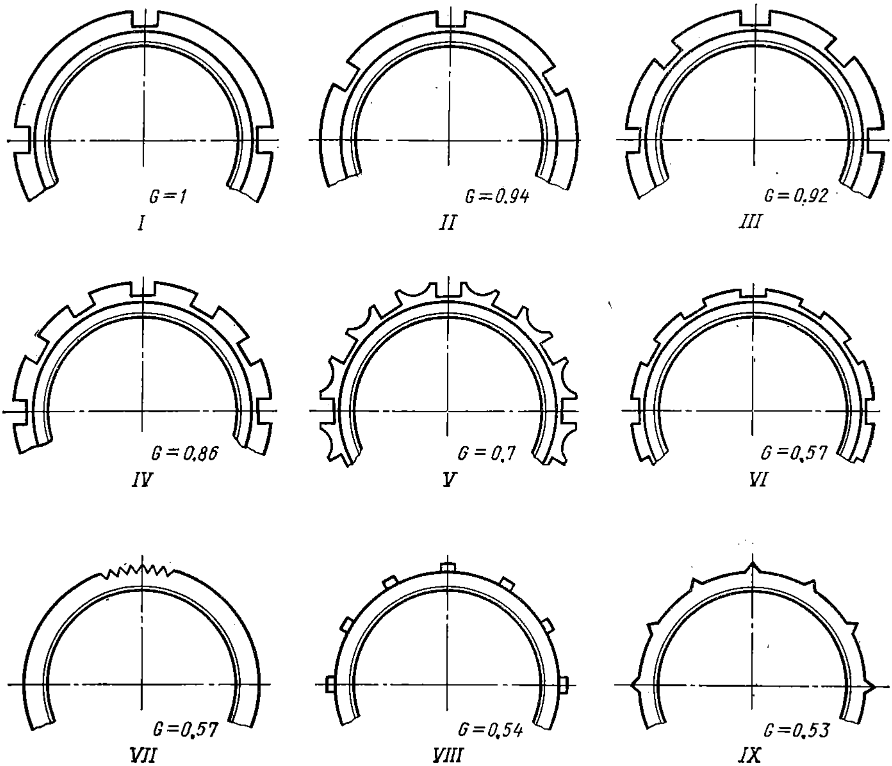


Fig. 115. Relative mass of circular nuts of different shape

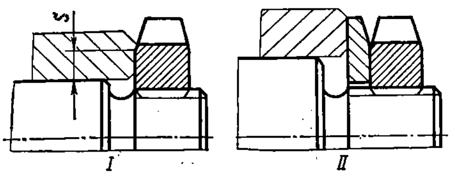


Fig. 116. Circular-nut assembly without washer (*I*) and with washer (*II*)

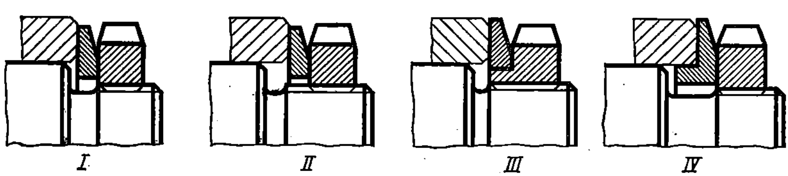


Fig. 117. Centring washers under circular nuts

I—no centring; *II*—on thread major diameter; *III*—on nut spigot; *IV*—in hub

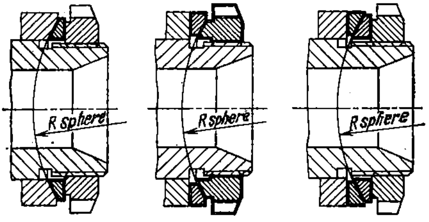


Fig. 118. Spherical-seat washers for circular nuts

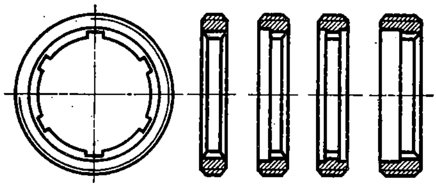


Fig. 119. Male threaded circular nuts with internal slots

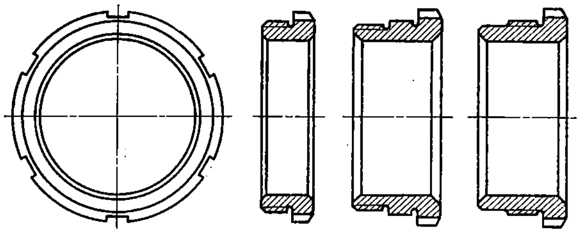


Fig. 120. Male threaded circular nuts with external slots

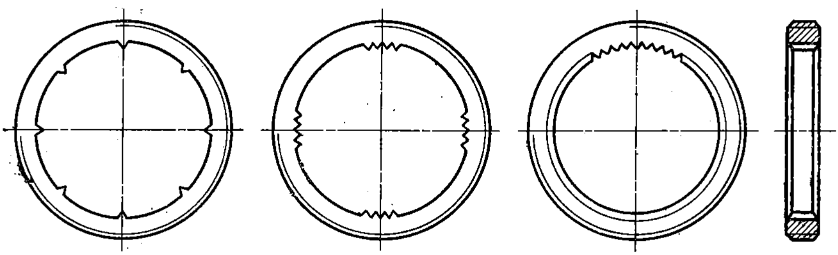


Fig. 121. Male threaded circular nuts with internal serrations

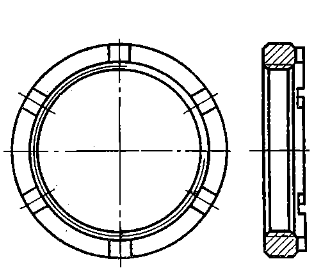


Fig. 122. Male threaded circular nuts with lateral slots

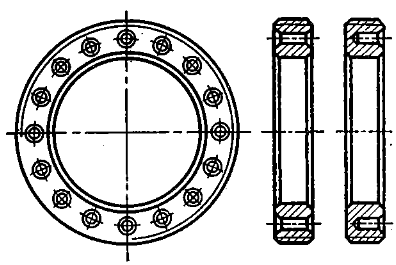


Fig. 123. Male threaded circular nuts with lateral holes

The nuts of Fig. 115, *VI-IX* are driven only with closed-end spanners.

When clamping parts with circular nuts, it is necessary that no less than 3/4 of the nut end-face height should bear against the part (dimension S in Fig. 116, *I*). If a shoulder on the shaft makes that impossible, use is made of a massive bearing washer (Fig. 116, *II*) placed between the nut and the part being clamped. It is important that the bearing washer should be centralized. Fig. 117, *I* shows an improper arrangement, where the washer can slip into the undercut

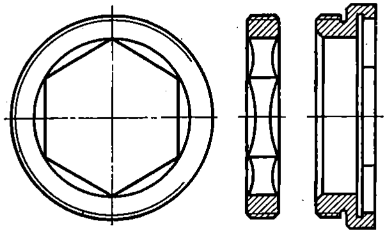


Fig. 124. Male threaded circular nuts with hexagon socket

behind the thread. Fig. 117, *II-IV* illustrates methods for centring the washers, the simplest being location from the thread major diameter (Fig. 117, *II*).

Where a uniform pressure on the component to be clamped is required, use is made of spherical seat bearing washers (Fig. 118). Other solutions to this problem are a strictly held squareness between the nut end face and the thread axis or, conversely, the use of a loose-fit thread with radial and axial clearances which allow a certain self-alignment of the nut on the shaft.

Figs. 119-124 present male threaded circular nuts of various shape and with various driving elements.

1.9. Miscellaneous Fasteners

1.9.1. "Captive" Nuts and Screws

In some cases, nuts, after they have been unscrewed by several turns, should be stopped so that they cannot be completely removed from the bolt. Such "captive" nuts are used for swing bolts and also for applications where the nut should be slackened by one-two revolutions to allow, for instance, positional adjustment of one component relative to another, etc.

Fig. 125, *I* and *II* shows the methods for keeping the nut on the bolt by peening or notching with a punch the bolt ends, and Fig. 125, *III*, by riveting a stop washer. A plain cylindrical portion

can be left on the end of a threaded shank (Fig. 125, *IV*) if the nut can be screwed on it from the opposite end.

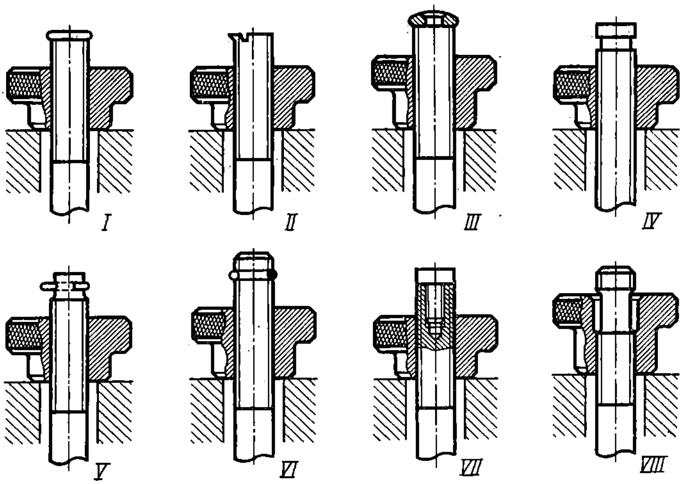


Fig. 125. Captive nuts. Methods for retaining

Of all the methods illustrated in Fig. 125, *V-VIII*, the simplest and most reliable method of nut retaining is that using a snap ring (Fig. 125, *VI*). In the construction of Fig. 125, *VIII*, the end of the bolt has a neck whose length is equal to that of the nut thread. As it is unscrewed, the nut is brought to the neck, and the threaded portion at the end to some extent prevents the nut from being completely detached.

Fig. 126 illustrates the use of "captive" nuts for fastening a lid to a housing with the aid of studs. The nut is retained on the lid by means of a snap ring. Shown in Fig. 126, *I*, is a single-stud application. Where the lid is fastened by several studs (Fig. 126, *II*), the snap ring should be spaced from the lid at distance *b* which is greater than length *a* of the nut end of the stud. That makes it possible to unscrew all the nuts irrespective of one another. Otherwise, a nut being unscrewed would jam with all the others still tightened.

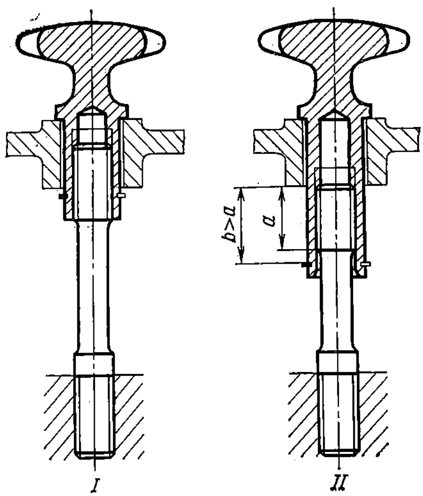


Fig. 126. Captive nuts as used to fasten a lid to a housing

Fig. 127 illustrates methods for retaining screws (also as applicable to fastening a lid to a housing). Fastening with a single screw is shown in Fig. 127, *I*. If several screws are to be used, it is necessary to observe the following rule: the retaining elements must be spaced

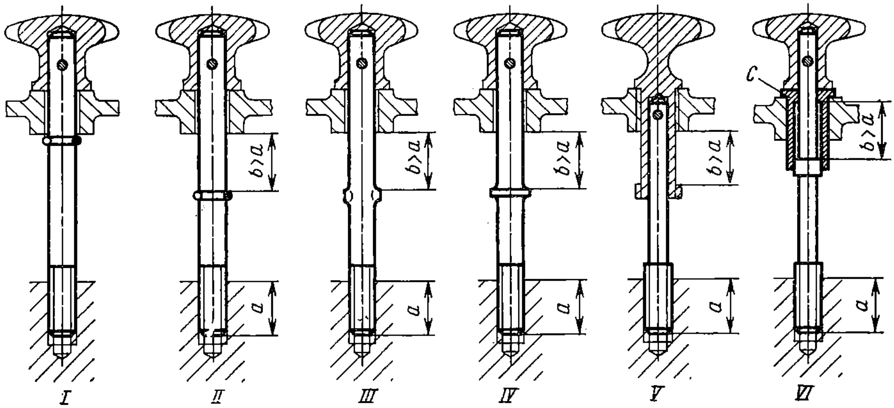


Fig. 127. Captive screws

I—fastening with a single screw kept by retaining ring; fastening with several screws kept by: *II*—retaining ring; *III*—wings pressed on the shank; *IV*—collar; *V*—thread; *VI*—collar which runs up against bushing *C*

from the lid at distance *b* which is greater than length *a* of engagement of the screw thread (Fig. 127, *II-VI*).

1.9.2. Swing Bolts

Swing bolts are used where quick disassembly of the joint is required, e.g. for fastening the covers of autoclaves (which is why bolts of this type are sometimes called "autoclave bolts").

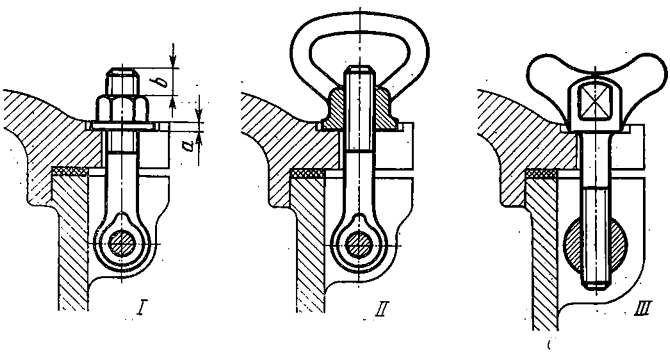


Fig. 128. Swing bolts

Some design rules must be observed in the application of swing bolts. The bearing surface on the lid for contact with that of the nut or bolt head should be counterbored by amount a (Fig. 128, *I-III*) sufficient to properly locate the tightened bolt and prevent it from spontaneous disengagement. The threaded end of the bolt should be provided with an additional portion, whose length b is determined so that the nut need not to be removed from the bolt when this is to be swung out.

The swing-bolt heads and the nuts are shaped so as to allow convenient turning by hand (Fig. 128, *II*, *III*), but tightening with a spanner must also be provided for. As shown in Fig. 128, *III* this requirement is met by introducing flats on the bolt head. The nut shown in Fig. 128, *II* can be tightened with a piece of rod passed through its eye.

1.9.3. Set Screws

These are mostly used for axial and angular location of components on shafts.

Shown in Fig. 129 are set screws with different driving elements and points which can form various combinations. Set screws fall into two main types: thrust type and seat type. Screws of the first type (Fig. 129, *I-V*) hold a component on a shaft by friction as the screw point bears against the shaft surface (Fig. 129, *XI*). Screws of the second type (Fig. 129, *VI-X*) provide positive location of the component, because the screw point enters a seat on the shaft (Fig. 129, *XII*).

The thrust-type screws have flat points (Fig. 129, *I*), oval points (Fig. 129, *II*), and cup points (Fig. 129, *III-V*) whose annular edges provide for better engagement of the component with the shaft. This type of set screw as a means for locking gradually goes out of use. Its main disadvantages are unreliable locking and misalignment of the component introduced in the process of screw tightening. In addition, screws offset relative to a diametral plane cause the component to get skew on the shaft.

Slotted set screws (Fig. 129, *I-III*) cannot be tightened hard, and it is practically impossible to lock them. Square and hexagon head screws (Fig. 129, *IV*, *V*) are free from these drawbacks, but they slacken gradually all the same due to deformation of the threads and screw points.

Set screws of the thrust type should not be used for axial location of parts. Only where the axial position of a component is not strictly specified, use may be made of such screws. It is self-evident that these screws cannot transmit torque, and therefore are used in combination with keys and other torque-transmitting means.

Fig. 130 gives examples of locating gear wheels on a shaft with the aid of set screws placed into the hubs of the wheels (Fig. 130, *I*, *II*) or into locating rings (Fig. 130, *III*).

Locating rings (Fig. 131) are now out-of-date, because they fail to provide the reliable location and positive locking of components, which is essential to operation under heavy loads. Furthermore, the assemblies prove to be very massive.

When a thrust-type set screw has to be used, a screw with a head and a cup point is the best choice (although it should be remembered

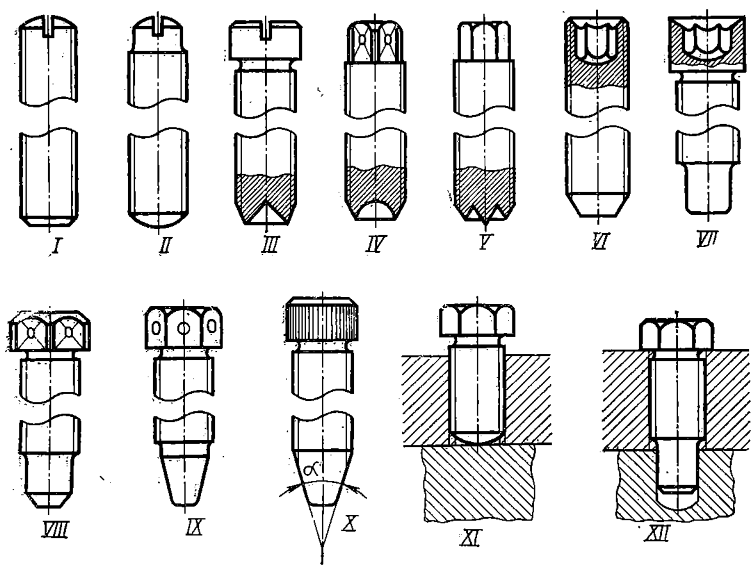


Fig. 129. Main types of set screws

that the cup edges may damage the surface of a shaft). Set screws must be heat treated to Rc 45-44, and the shaft surface hardness must not exceed Rc 30-35.

The disposition of a set screw with reference to an associated key is of importance. The most effective of all the dispositions presented in Fig. 132,*I-IV* is that shown at *IV*, where the screw is disposed at an angle $\alpha = 30\text{-}45°$ to the key's axis of symmetry. Here, the tightening of the screw produces some pressure of the hub's keyseat upon the bearing side of the key (the direction of the torque applied to the hub is indicated in Fig. 132, *IV* by an arrow).

When torque is applied in the opposite direction, the screw disposition should be accordingly reverse.

The seat-type set screws are used to fix parts against axial and angular displacements, and also to transmit light torques.

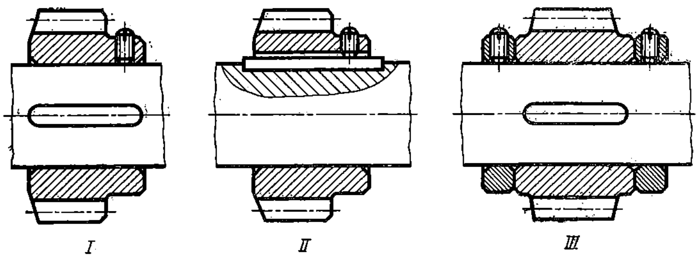


Fig. 130. Set screws as used for locking gear wheels
I—screw bears down against shaft; *II*—screw bears down against key; *III*—locating ring with set screws bearing down against shaft

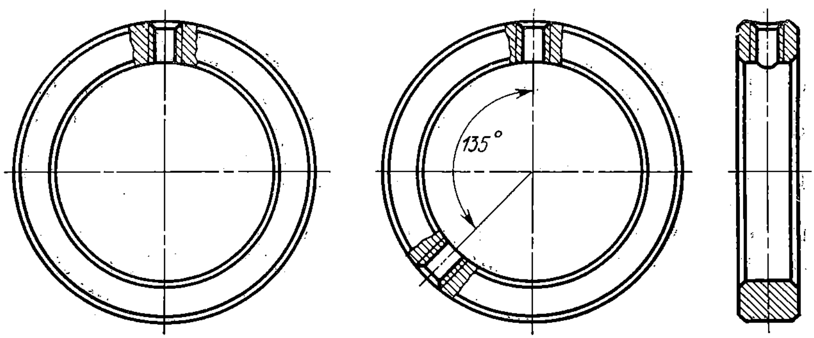


Fig. 131. Locating rings

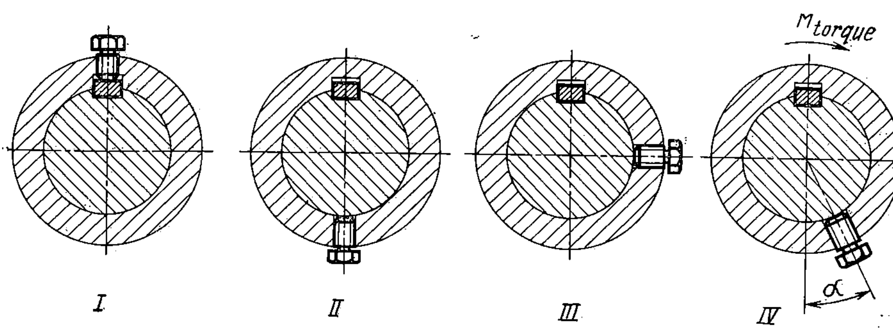


Fig. 132. Methods of positioning set screws in keyed assemblies

Set screws with dog points (Fig. 129, *VII*, *VIII*) are inserted mostly into holes previously drilled in shafts. When axial adjustment of a part is required during assembly, seat drilling is done in situ through the tapped hole in the part. Depending on operating conditions, the screw point is inserted into the seat with the application of a free or close fit.

It must be taken into account, that drilling and, moreover, reaming (if a close fit is required) of the seat in the course of assembly involves difficulties. What is more, the chips may get into the assembled unit. It is often that the unit must be disassembled and cleaned in order to get rid of the chips.

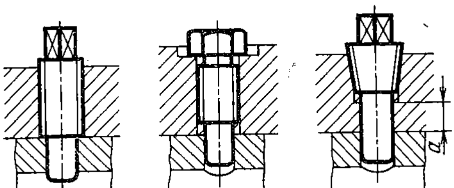


Fig. 133. Set screws with dog point as used for locking inner bushings

I and *II*—incorrect and correct arrangements, respectively; *III*—assembly for heavy loads

When using set screws with dog points, especially for securing thin-walled parts (e.g. bushings), the screw should be set to bear on its head (Fig. 133, *II*). Otherwise, the screw will press by its threaded portion upon the part (Fig. 133, *I*) and deform it.

The ability of an assembly to withstand shearing loads is increased if the tapped hole in the hub is provided with a plain portion *a* (Fig. 133, *III*) equal in diameter to the seat in the shaft, both these holes are reamed in one setup, and the point of the set screw is inserted in these holes by a close fit. The thread of the screw is thereby relieved from shearing stress, and the locking proves to be more reliable.

The most effective assembly is provided by set screws with tapered points (see Fig. 129, *IX*, *X*). The included angle α of the taper normally ranges from 20 to 30°.

The merit of such screws is that they do not require locking; a point taper angle of 15 to 20° will secure the screw from loosening.

Examples of assemblies with set screws having tapered points are shown in Fig. 134, *I*-*V*. Here, the nearer the taper base to the mating surfaces of the assembled parts, the stronger is the joint. For this reason, the taper angle of the seat is well to take somewhat smaller (by 30′ to 1°) than that of the screw point, so that the taper base comes in contact with the mouth of the seat (Fig. 134, *III*).

Still greater strength is obtained where the screw point is inserted into the seat drilled and reamed at once in both assembled elements (Fig. 134, *IV*). Sometimes the tapered point is inserted into a cylindrical hole (Fig. 134, *V*). This facilitates manufacture, and the assembly proves to be strong enough since the screw point forms a seat for itself by breaking the hole edges.

Fig. 135 shows the locking of the hub of a lever on a shaft by a set screw with a tapered point. Here, a double task is effectively carried out: torque is transmitted from the lever to the shaft, and

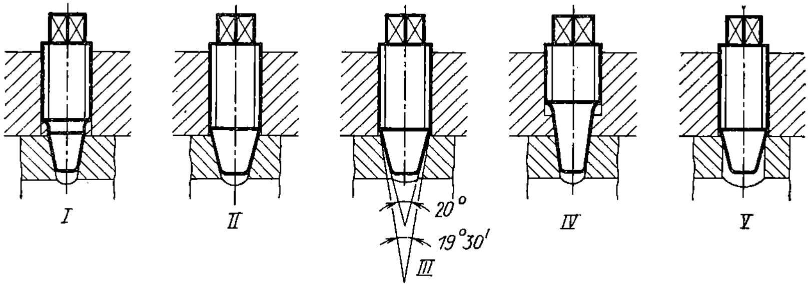


Fig. 134. Set screws with tapered points used for locking inserted bushings

the shaft is locked axially through the hub whose end faces are captured between the housing walls.

Set screws with dog points are often employed where the hub must be axially fixed and, at the same time, left free to rotate on the

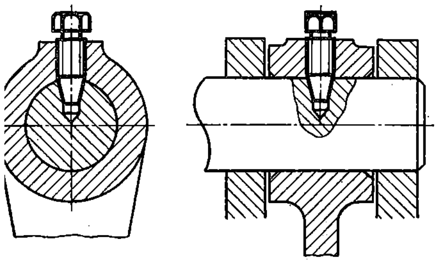


Fig. 135. Axial and angular locking of lever on shaft with tapered-point set screw

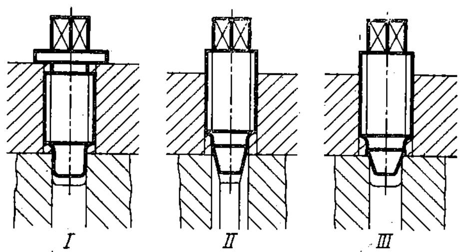


Fig. 136. Set screws used for axial and angular locking of hubs on shaft

shaft. To effect this, a circular groove is provided on the shaft, and the screw point is inserted therein (Fig. 136, *I*).

Set screws with tapered points are used where the axial and angular location of the hub must be ensured, yet its angular adjustment should be allowed for. To this end, the circular groove on the shaft is shaped according to the profile of the screw point (Fig. 136, *II*). After it has been set on the shaft in the proper angular position, the hub is fixed by tightening the screw. With the point taper angle ranging from 15 to 20°, a self-holding effect is achieved.

Where the hub is to be set in the required angular position only once, a circular groove of rectangular cross-section may be used (Fig. 136, *III*).

1.10. Pins

1.10.1. Fastening Pins

Pins as fasteners have found limited application. They are used for assemblies subject to light loads, mainly for fastening components on shafts (Fig. 137) and axles in housings (Fig. 138, *I*). The drawbacks to this kind of fastening are the weakening of the shaft by a hole drilled for the pin, low shear strength, lack of tightening,

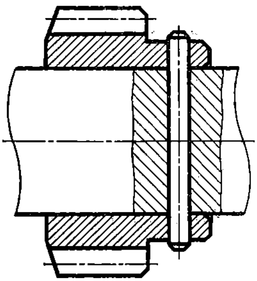


Fig. 137. Fastening gear wheel on shaft with cylindrical pins

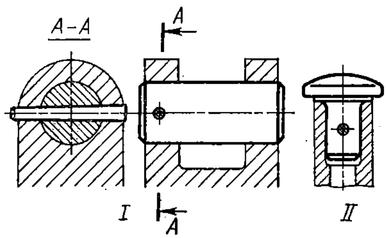


Fig. 138. Fastening parts with taper pins

cumbersome assembly and disassembly, difficulties in manufacture (the hole for the pin must be drilled with the assembled components in position).

Pins cannot be used for fastening parts hardened in excess of the limit that determines machinability of the material (e.g. over Rc 30). Thus, for instance, an anvil-type part (Fig. 138, *II*) hardened to Rc 60-62, which is to be assembled with a tubular part, cannot be drilled and reamed together with this part. In such cases, the spots to be drilled must be left unhardened; subjected to hardening (Rc 60-62) should be only the working surfaces, for which purpose use may be made of carburizing, induction hardening and some other methods.

The use of pins for taper assemblies is impermissible, even if both components are drilled together for pin insertion. During repeated assemblings, the male and female tapers change their relative position due to variation in the tightening force and wear of the mating surfaces. After tightening, insertion of the pin may prove impossible due to the displaced pin holes in the mating parts. The pin inserted prior to tightening may be sheared off after the joint has been tightened.

Fastening pins are divided into two main types: cylindrical and taper pins.

Cylindrical Pins. Shown in Fig. 139 are various cylindrical pins: *I*—plain; *II*—with a slow-taper guiding end; *III*—with ends pre-

pared for staking; *IV*—with slotted ends for locking with a retaining ring; *V*—hollow pin; *VI*—hollow self-locking pin.

Pins are made of carbon steels with a 0.45-0.5% carbon content and heat treated to Rc 40-45. Holes for pins are drilled and reamed in the assembled parts placed in position. Pins are installed in the holes with interference fits.

Holes for pins should be drilled through; otherwise, the pin will be impossible to remove for disassembly. Blind pin holes are per-

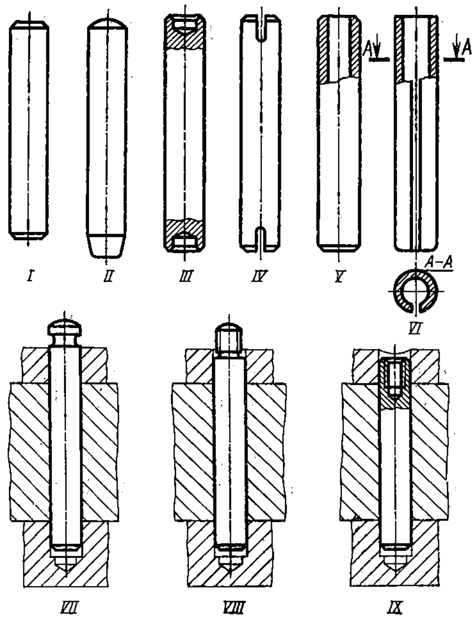


Fig. 139. Forms of cylindrical pins

missible for permanent joints only. However, if such joints still have to be dismantled, the pins must have elements provided for this purpose (Fig. 139, *VII-IX*).

Despite the interference fits, pins must be locked in position. Methods for locking permanent joints with pins are illustrated in Fig. 140. The ends of unhardened pins are closed (Fig. 140, *I*, *II*). In the case of hardened pins, their ends are closed by the surrounding material driven by caulking or notching with a punch (Fig. 140, *III*).

Figs. 141 and 142 illustrate methods for locking pins in detachable joints. The pin of Fig. 141 is locked by a retaining ring, which is placed on a groove made in the hub. Gap *a* between the ring ends should be smaller than the pin diameter, so that the pin will not

slip out of the hole if by chance the gap happens to be at one of the pin ends.

The pin in Fig. 142 is locked by a retaining ring placed into slots which are made in the pin ends. Here, as distinct from the design

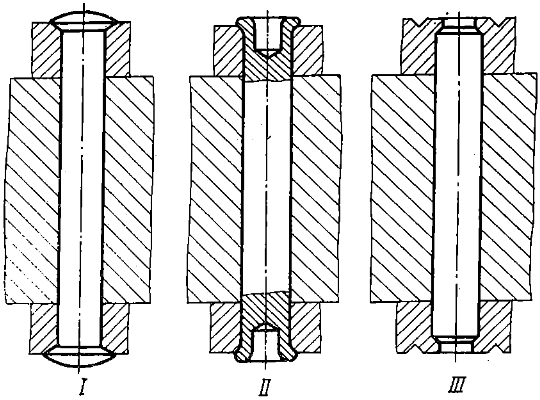


Fig. 140. Methods for locking cylindrical pins

of Fig. 141, the hub has no groove which weakens it. In assembly, the pin should be so oriented that the slots are in a plane square to the shaft axis.

The described locking methods can be used only where the frequency of rotation of the shaft is low. When it is high, the retaining

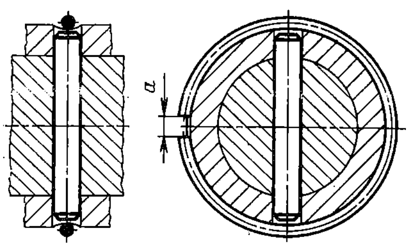


Fig. 141. Pin locked with round retaining ring

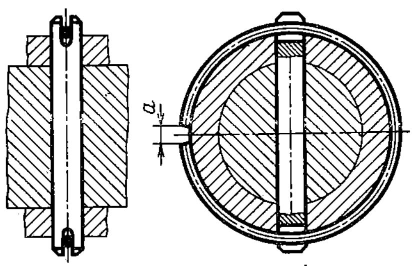


Fig. 142. Retaining ring locks pin as it engages in slots on pin ends

ring may open under centrifugal forces.

In this case, a rather primitive method of locking is sometimes resorted to; that is, the ends of the pin are locked by binding with a wire placed in a groove on the hub.

Grooved Pins. These pins (Fig. 143, *I-VI*) are used for non-critical applications. They are made of steel, hardened to Rc 40-45, ground,

and then subjected to a treatment, whereby grooves of triangular profile are formed thereon. The edges of the grooves project beyond the pin's cylinder as shown in the enlarged sectional views of Fig. 143. The pins are forcibly driven into drilled or reamed holes, so that the edges cut into the walls, providing more or less reliable locking of the pin.

The disposition of grooves along the pin length depends on the conditions of the joint's assembly and operation. Fig. 144, *I*, *II* shows grooved pins with heads for driving with a hammer.

Grooved pins can be driven into plastic materials only; they are unfit for brittle materials. Some assemblies with grooved pins are shown in Fig. 145, *I*, *II*. In some cases, larger parts (pivot pins, axles, etc.) to be assembled with the application of forced fits are provided with grooves (Fig. 145, *III*).

Use is also made of self-locking pins with knurled collars (Fig. 146, *I-V*).

Parts of soft materials (wood, plastics, etc.) are assembled with the use of pins having spiral ridges (Fig. 147).

Taper Pins. These are used in critical applications. The conical shape provides a tighter fit; the pins require simpler locking arrangements, because here they should be secured against slipping out at one end only.

However, taper pins and holes are more difficult to make.

When setting taper pins, it is easy to cause considerable tensile and bearing stresses in the walls of the holes. For this reason, the pins are driven by a strike with a calibrated force, or on a press with controlled pressure. Taper pins are not recommended for use in soft materials (such as aluminium and magnesium alloys).

Fig. 148, *I-VIII* shows taper pins in various forms. Pins with external or internal threads for extraction (Fig. 148, *VII*, *VIII*) are used for blind holes and also as locating elements.

Taper pins feature a standard taper of 1 : 50. Despite the fact that this taper provides self-holding, taper pins are locked against falling out with additional means.

Some methods for locking are illustrated in Figs. 149-152. Pins of soft steel are locked by riveting or by spreading the halves of the split end as shown in Fig. 149, *I*, thereby forming a permanent joint. Use is also made of locking by a cotter pin (Fig. 149, *II*), or of tightening by the nut with subsequent locking by a cotter pin (Fig. 149, *III*).

Shown in Fig. 150-152 are common methods for locking taper pins with retaining rings. The pin of Fig. 150 is locked by a ring placed in the pin slots and the groove in the hub. Gap *a* between the ring ends must be smaller than the pin's small-end diameter. The drawback of this method is the necessity to insert the pin in a specified position where its slots coincide with the groove on the hub.

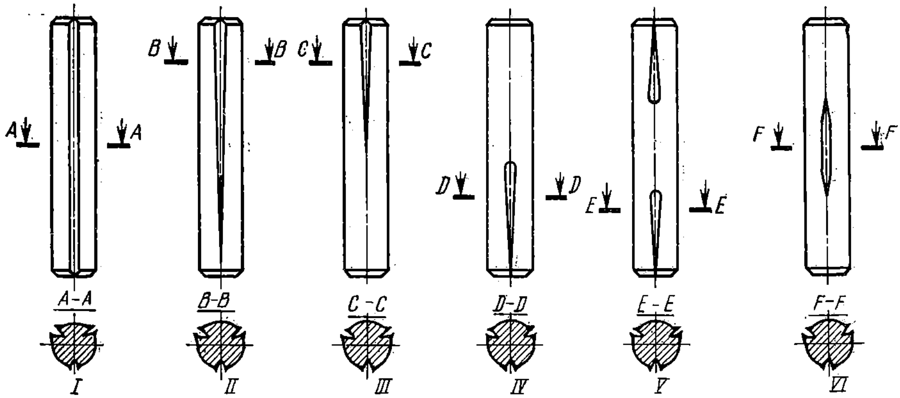


Fig. 143. Grooved pins

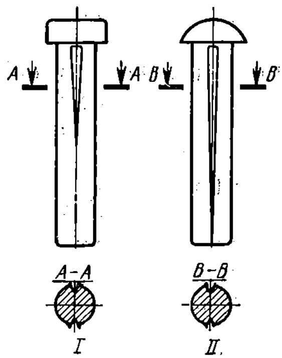


Fig. 144. Grooved pins with heads

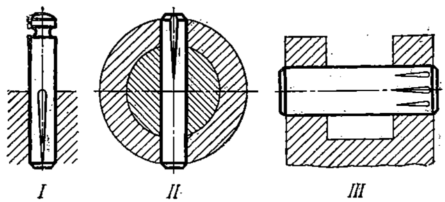


Fig. 145. Grooved-pin assemblies
I—spring-anchoring pin; *II*—fastening hub on shaft; *III*—fastening pivot pin

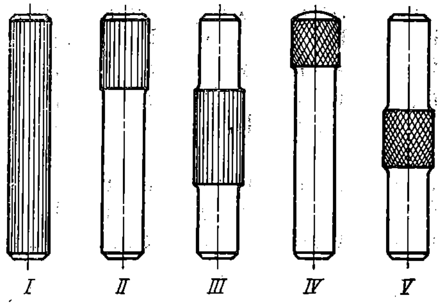


Fig. 146. Knurled pins

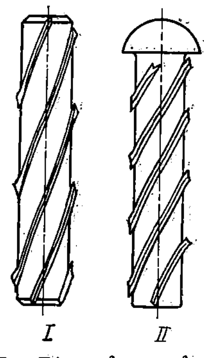


Fig. 147. Pins for soft materials

A better method is shown in Fig. 151. Here, the pin is so installed that its large end is sunk in the hole and the small end comes outside. The pin is locked by a snap ring which, in turn, is secured against rotation by the jutting end of the pin. Where the pin large end cannot

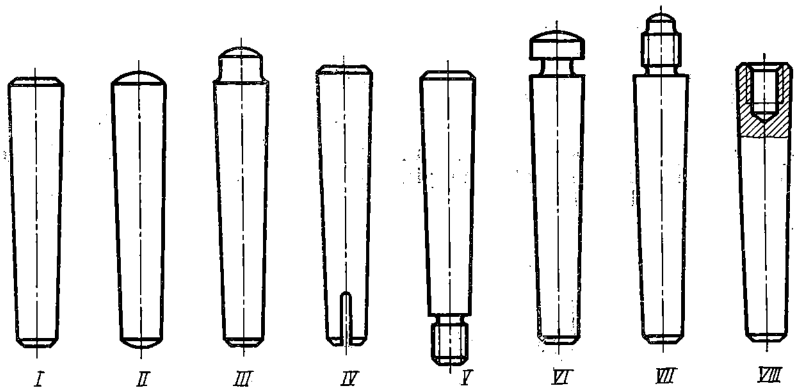


Fig. 148. Taper pins

be sunk into the hole, which is the case with thin-walled hubs, use is made of the arrangement shown in Fig. 152. Here, the retaining

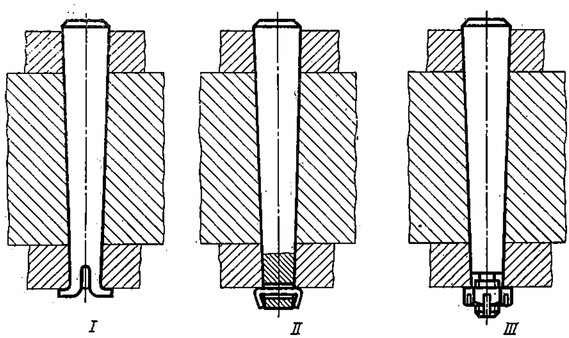


Fig. 149. Methods for locking taper pins

ring has a loop-shaped portion which captures the jutting end of the pin, so that the pin is locked in position and the ring is fixed against rotation.

The use of retaining rings for pin locking is limited to applications where the shafts rotate at slow speeds.

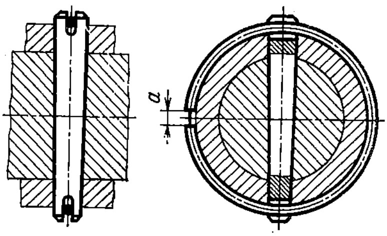


Fig. 150. Taper pin locked **with round** retaining ring inserted **into the pin** end slots

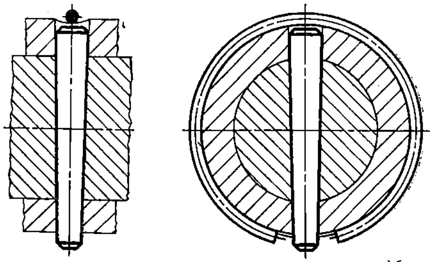


Fig. 151. Taper pin locked **with round** retaining ring

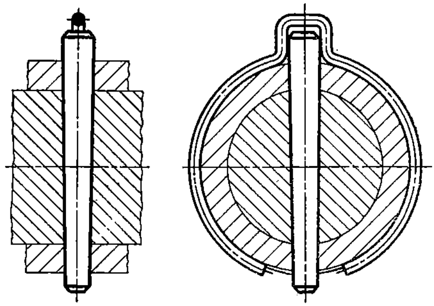


Fig. 152. Taper **pin** locked by retaining ring having a loop for pin end

1.10.2. Locating Pins

Locating (or dowel) pins are used where one part must be positionally fixed relative to another (e.g. a bearing cap located on a bearing housing) and where shear loads acting at the junction of two parts (e.g. in transmitting torque from one shaft to another by the flanges of a permanent coupling) must be withstood.

Two types of locating pins are employed: cylindrical and removable taper pins.

Cylindrical dowel pins (Fig. 153) are usually rigidly set in one of the assembled components, the projecting end of the pin being inserted into a bore in the other component by a sliding fit or a light drive fit.

The shape of a pin is important for the performance and useful life of the assembly. The simplest cylindrical pin with a chamfer of 45° (Fig. 153, *I*) is the least satisfactory, because the chamfer edges impair the walls of the bores as the pin is initially driven, or the detachable part is set, in place.

Although chamfers of 10 to 20° (Fig. 153, *II*) are better, the same drawback to a lesser degree can also be found here. The best effect is achieved when the pin end edges (at least those entering the detachable part) are rounded (Fig. 153, *III*).

The optimum shape is shown in Fig. 153, *IV*. Here, the profile of the pin locating end is outlined by a curve of variable radius which smoothly joins the generator of the pin cylinder. This form is widely used for pins assembled to light-alloy components. Such pins are somewhat harder to produce, but they provide for convenient mounting and long service life of the assembly.

Pin holes in assembled parts must always be chamfered. The chamfers provide guidance for pins during their driving; with parts of plastic metals, they prevent the raising of a lip around the hole. The holes in detachable parts are chamfered to facilitate assembly.

Pin holes in parts of soft materials, e.g. plastics, are provided with bushings for the pin locating end (Fig. 153, *V*). The bushings are secured by means of a thread, interference fits, etc.

Locating pins are made of high-carbon steels for general applications and of alloy steels with hardening to Rc 50-60 for critical high-load applications. The working surfaces of pins are machined to within the limits of the 2nd tolerance grade and to a surface finish of 0.32-0.63 μm Ra.

Pins are inserted in blind holes usually with interference fits. The smaller the pin diameter and the softer the material of the component, the greater is the amount of interference to be applied. Nevertheless, pins are often additionally secured against slipping out (Fig. 160), which may occur as a result of accidental loosening of the pin.

The depth of pin setting l (Fig. 154) depends on the housing material and pin diameter d.

Material	l
Steel, high-grade and malleable cast iron, bronze	about $2d$
Grey cast iron	$(2\text{-}2.5)\,d$
Aluminium, magnesium, and zinc alloys	$(2.5\text{-}3)\,d$
Plastics (without inserts)	$(3\text{-}3.5)\,d$

For small pins (under 3-4 mm in diameter), the above-mentioned values should be increased 1.5-2 times.

Height h of the locating end of the pin (Fig. 154) is taken to be no less than $(1.5\text{-}2.5)\,d$ (greater values are used for soft materials). Small and medium-size parts are dowelled with pins of 4 to 10 mm in diameter, and large parts, with pins of 10 to 20 mm in diameter. Large-diameter pins are often made hollow (Fig. 155) to reduce their mass.

Pins are installed in blind holes by one of the three methods: they are brought to bear against the end of the hole's reamed portion (Fig. 156, *I*), against the bottom of the hole (Fig. 156, *II*); and

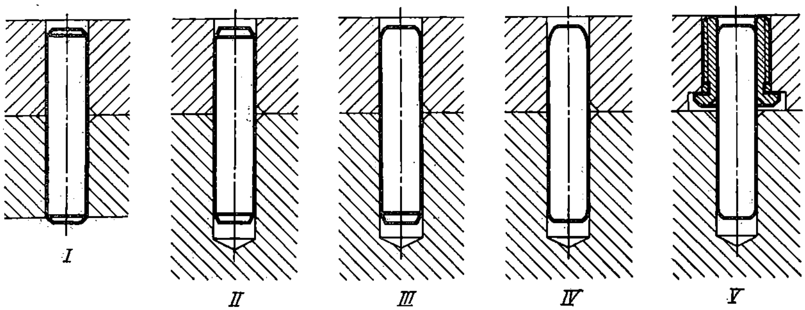


Fig. 153. Cylindrical dowel pins

they are driven by means of a special tool (Fig. 156, *III*) which provides the specified height of the pin locating end. The latter method is the most preferable; it is also employed for driving pins in through holes.

The depth of blind holes must allow for the reamer chamfer. Distance *a* (Fig. 156, *III*) between the bottom and the reamed part of the hole should be at least $0.6d$ for hand reaming and $1.5d$ for machine reaming.

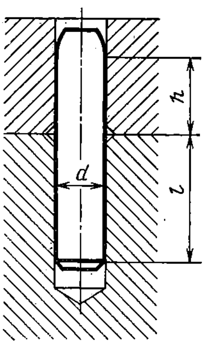


Fig. 154. Dimensional relationships for cylindrical dowel pins

When driving pins in blind holes, provision must be made for letting the air escape therefrom in order to prevent the hole walls from bursting (especially in parts of light alloys). For this, air ducts are made either in the walls of the pin hole (Fig. 157, *I*, *II*) or in large pins (Fig. 157, *III*, *IV*).

Three methods of setting pins in assembled parts are used. The first method (Fig. 158, *I*) involves the application of the basic shaft system of fits. The pin is made plain, of the same diameter throughout its length; the hole wherein the pin is to be driven is reamed to provide a press fit; the hole in the detachable part is reamed to provide a sliding or close fit, depending on requirements. This is the most commonly used method.

The second method (Fig. 158, *II*) makes use of the basic hole system of fits. Both holes are reamed to the limits of the 2nd tolerance grade, one end of the pin is machined to provide a press fit, and the other,

a sliding or close fit. As a result, the pin has two steps (which is undesirable for manufacturing reasons).

According to the third method (158, *III*), the pin is inserted into the holes by a close fit in the basic hole system. Both the pin and the holes are plain. Here, the pin must be locked in the base.

In single-piece and small-lot production, at least one of the two holes (better both) in assembled parts is arranged as a through hole,

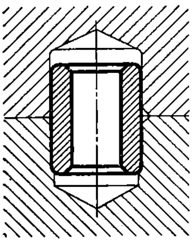


Fig. 155. Hollow dowel pins

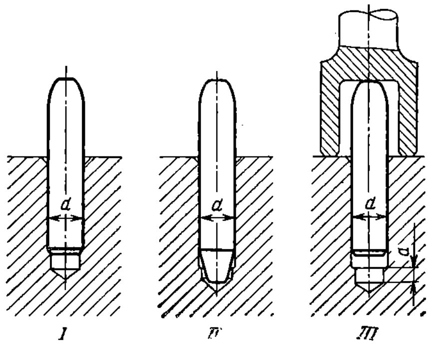


Fig. 156. Setting cylindrical dowel pins in blind holes

so that both holes can be drilled and reamed together (Fig. 159, *I*, *II*). Here, the basic hole system of fits should be applied (because otherwise an additional reaming of the hole for the pin locating end may introduce misalignment of the holes).

In large-lot production, pin holes in both assembled components are machined with the aid of special tooling (mirror-image jigs)

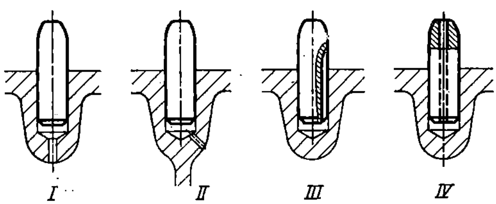


Fig. 157. Eliminating air compression when driving pins into blind holes

which ensures alignment of the holes with a high degree of accuracy. Here, the reaming of holes in both parts at once would complicate the manufacturing process. The use of special tooling makes it possible to machine accurately aligned blind holes (Fig. 159, *III*, *IV*). However, through holes are always preferable for higher accuracy and greater output of production.

Fig. 160 illustrates methods for locking pins against falling out, which may occur if the pin loosens in the hole (especially in parts of soft metals). Setting pins in two blind holes (Fig. 160, *I*), holding

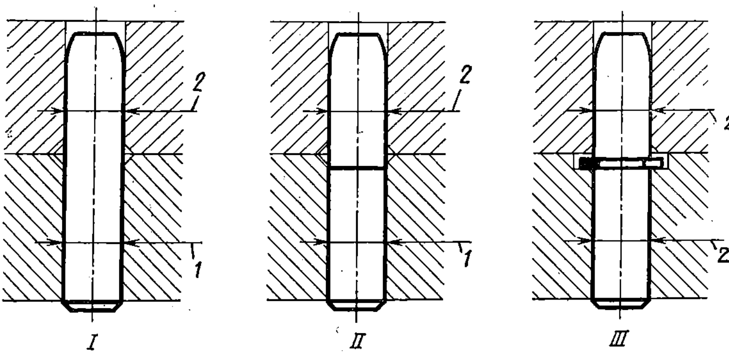


Fig. 158. Fits for cylindrical dowel pins
1—press fit; 2—sliding or close fit

by a retaining ring in a detachable part (Fig. 160, *II*) or at the interface of two assembled parts (Fig. 160, *III*) provide a fairly reliable pin locking.

What is more difficult is to prevent the pin from getting lost during disassembly. Locking by a single retaining ring (Fig. 160, *IV*) is not always possible for dimensional reasons; locking by two retaining rings (Fig. 160, *V*) is possible only with a through pin hole in the base. Locking by forcing the base metal around the hole into a groove on the pin (Fig. 160, *VI*) is applicable only to plastic metals.

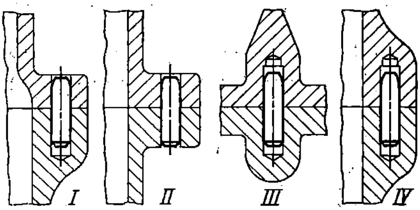


Fig. 159. Methods of dowelling

Fig. 161 shows an arrangement whereby the pin is secured by a large-diameter washer placed under the nut on the adjacent stud.

Definite rules proven by engineering practice must be observed in setting dowel pins. The pin should be sunk in the mating hole of the detachable part (Fig. 162, *II*). The end of the pin must not jut above the surface of the part (Fig. 162, *I*), since the pin may be damaged by an accidental impact or loosened in its seat. Where the flange is not thick enough for a fully sunk pin, provision is made for local bosses (Fig. 162, *III*).

Locating pins should be disposed near fasteners, e.g. bolts, studs, etc. If the parts to be assembled have no aligning elements (e.g.

centring spigots, recesses, collars), two locating pins must be used for their alignment. A greater number of pins is unreasonable unless the joint is subjected to heavy shearing loads. Where the assembled

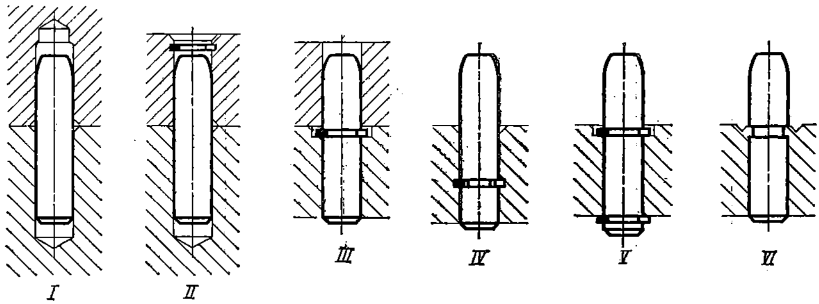


Fig. 160. Locking cylindrical dowel pins against falling out

parts have a centring feature, a single pin will be quite sufficient to provide their definite angular location relative to each other. Again, with heavy shearing loads more dowel pins may be used.

Locating pins should be spaced from a component's axis of symmetry and from each other as far as possible.

Fig. 163 presents examples of the proper and the improper location of dowel pins on a lid-type part (the pin holes are shown as half-shaded circles).

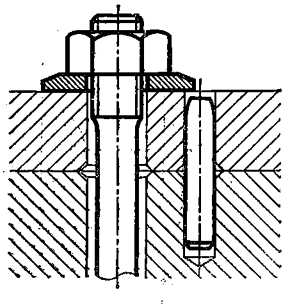


Fig. 161. Dowel pin locked with washer

In the design of Fig. 163, *I*, location of the pins is incorrect because they are placed far from the bolt holes. In the design of Fig. 163, *II*, the pins are located near the bolt holes, but the drawback is in the small distance between the pins. They fail to fix the lid for reliable alignment, because shearing loads (due, for instance, to working forces applied to the lid central boss) give rise to high stresses in the pin joints. Location of the pins can be improved by placing them farther from each other, as shown in Fig. 163, *III*. The most advantageous location is shown in Fig. 163, *IV*, where the pins are spaced as far apart as possible.

Taper dowel pins provide for greater accuracy of alignment than cylindrical pins. This accuracy remains practically the same after many dismantling operations and wear, since a close fit of the pin is recovered by driving it deeper each time. Another advantage of these pins is that they are relatively easy to remove, which makes it possible to replace damaged pins and thereby simplify assembly

and disassembly. Taper pin assemblies are substantially more difficult in manufacture than those with cylindrical pins. Here, pin holes in the components being assembled are drilled, core-drilled, and reamed in situ.

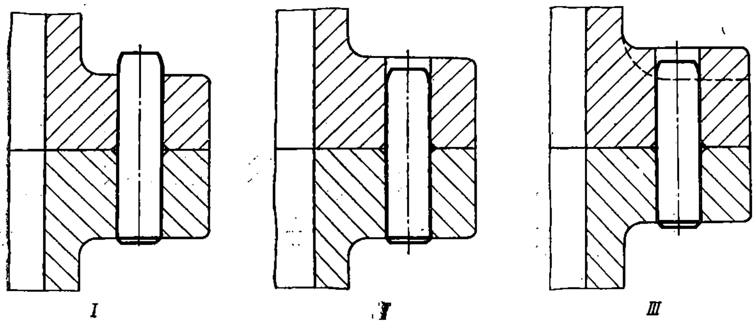


Fig. 162. Setting cylindrical dowel pins

The pins are made of steel and hardened. Their standard taper is 1 : 50. They are driven into the holes with a calibrated force.

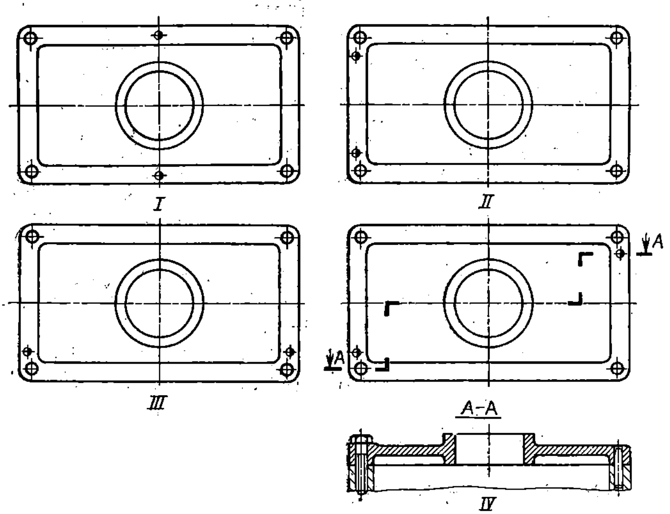


Fig. 163. Location of dowel pins on dowelled parts

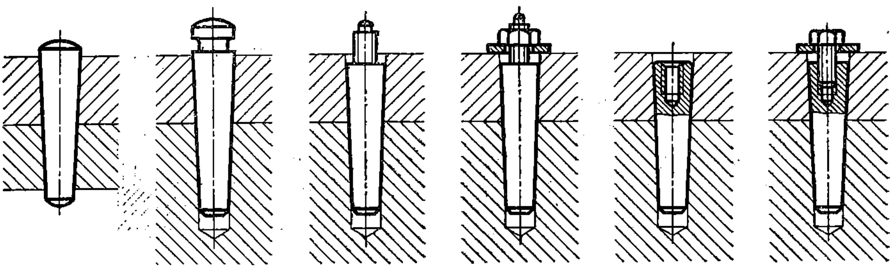
Taper pins are not used for dowelling light-alloy components for fear of crushing the walls of pin holes during pin driving.

The main types of taper dowel pins are shown in Fig. 164. The pin of Fig. 164, I is used only for permanent joints or for through-

hole assemblies where they can be extracted by striking from the underside.

In detachable joints and blind-hole applications, the use of pins with features designed for pulling-out is indispensable.

The simplest element used for pin removal is a circular groove on the pin end (Fig. 164, *II*) to receive the jaws of a puller. Some-



I *II* *III* *IV* *V* *VI*

Fig. 164. Taper dowel pins

what more convenient are an external (Fig. 164, *III*) and internal (Fig. 164, *V*) threaded features. Such pins are extracted with the aid of a nut (Fig. 164, *IV*) or a bolt (Fig. 164, *VI*), which bear through a washer against the surface of the detachable part. The nut or bolt is turned a few revolutions only, which is enough to disengage the pin from its seat, and then it is removed by hand.

Unlike cylindrical dowel pins which allow for use of sealing gaskets at the interface of assembled parts, taper pins can properly function only in 'metal-to-metal' joints.

Taper dowel pins are employed in assemblies where the principal requirement is the accuracy of location.

Locating pins, whose axes lie in the plane of the joint faces of assembled parts (Fig. 165), are sometimes used for alignment purposes.

The locking of such pins by retaining rings (Fig. 165, *I*) cannot ensure that the pin will not slip out of its place during assembly and disassembly. A more reliable locking is achieved by fastening the pin with screws (Fig. 165, *II*) to one of the assembled parts or by squeezing the metal of the part into recesses in the pin body (165, *III*). The latter method is applicable only to parts made of adequately plastic metals.

Pin holes are drilled from the sides and reamed in the two assembled parts at once. The virtue of the resulting joint is a large cross section of the pin loaded in shear. The disadvantage is that the acting shearing loads give rise to forces perpendicular to the joint

faces of the assembled parts, thereby additionally loading the fastening bolts.

These pins are used only for metal-to-metal joints. They locate components only in the direction perpendicular to the pin axis. Where location in all directions is required, use is made of several

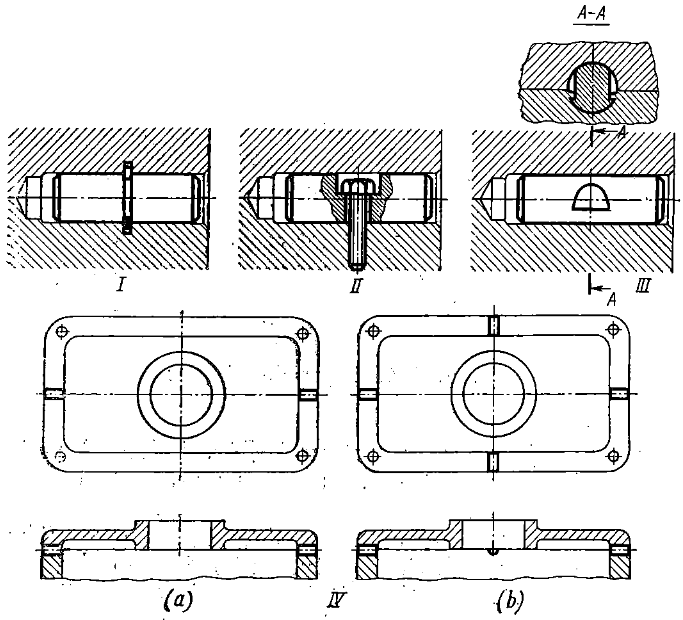


Fig. 165. Setting dowel pins along the joint face (*I-III*) and their arrangement (*IV*). Alignment in one direction (*a*) and two (*b*) directions

pins placed perpendicular to one another (Fig. 165, *IV*).

The accuracy of location by this type of pin is much lower than that by conventional cylindrical and taper dowel pins.

1.11. Locating Elements for High-Load Applications

These elements are used in assemblies subjected to heavy shearing loads to carry these loads and relieve bending of fasteners (i.e. in permanent couplings of shafts transmitting high torques). These elements fall into three categories: locating pins, dowel bolts, and locating bushings in combination with bolts.

In its simplest form, this kind of *locating pin* is a cylindrical pin set into one of the assembled components with a press fit, and into the other with a close fit (Fig. 166, *I*).

The dimension and number of such pins are determined by a calculation of their shearing and bearing stresses due to a force acting on the joint (without regard to the forces of friction that arise from tightening the fasteners). The pins are located between the bolts which are placed in regular clearance holes.

In order to prevent the crushing and distortion of pin holes in heavily loaded assemblies, especially those subjected to impulse

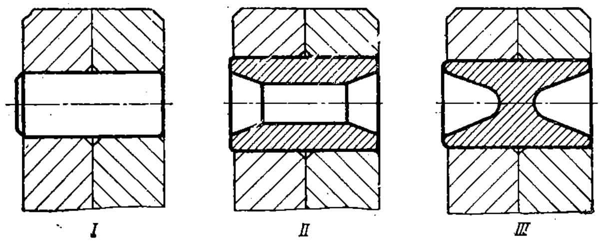


Fig. 166. Locating pins for high-load applications

or alternating loads, use is made of large-diameter pins. These are often made hollow for reduced mass (Fig. 166, *II*). The cross-sectional area of the pin in the plane of action of shearing forces can eb

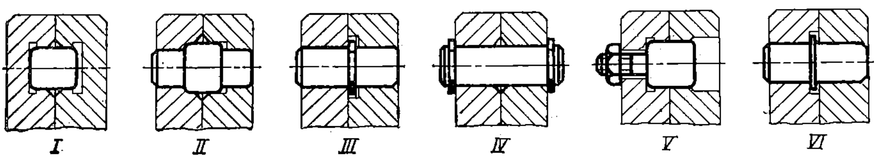


Fig. 167. Methods for locking locating pins in high-load assemblies

increased as shown in Fig. 166, *III*. The cross-sectional area receding toward the ends of the pin provides a uniform distribution of the bearing stress along the pin.

Locating pins are made of carbon steels with a 0.45-0.8% carbon content or of alloy steels for critical applications, and heat treated to Rc 45-55.

Although the pins are set in one of the components by an interference fit, they are additionally locked whenever necessary, since loading may loosen a pin in its seat and cause its disengagement, which sometimes occurs in practice. Methods for locking are illustrated in Fig. 167.

The designs of Fig. 167, *I*, *II* are inadequate from the manufacturing standpoint, as the machining of accurate blind holes is difficult. Locking by snap rings disposed at the interface of assembled parts

(Fig. 167, *III*) weakens the pin and results in its bending instead of direct shear loading.

The arrangement of Fig. 167, *IV*, where the pin is locked by snap rings at both ends, is satisfactory in terms of producibility and strength, but is unwieldy in maintenance, for one of the rings must

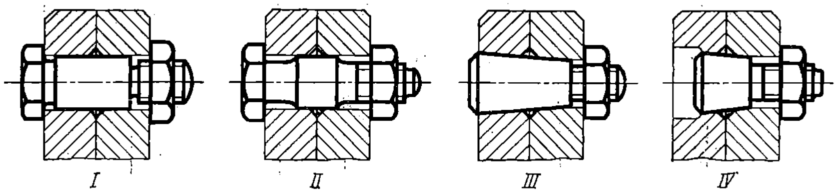


Fig. 168. Dowel bolts

be removed each time the assembly is to be taken apart. The most advantageous designs are those where the press-fit end of the pin is locked by a nut (Fig. 167, *V*) and where the pin has a collar located at the junction of the mating surfaces of the assembled parts (Fig. 167, *VI*).

Cylindrical dowel bolts (Fig. 168, *I*, *II*) allow the parts being assembled to be fastened together. This is an advantage over cy-

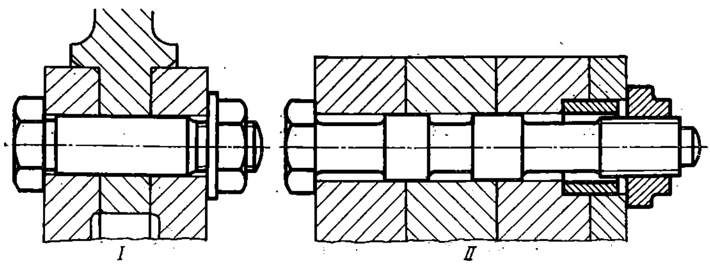


Fig. 169. Dowel bolts for fastening several components

lindrical locating pins. Heavily loaded units can be assembled entirely with the aid of dowel bolts. Actually, several dowel bolts are used in combination with plain bolts in order to reduce the volume of precision machining. The required number of dowel bolts is found by strength calculations.

Dowel bolts are inserted into associated holes with locational transition fits. To facilitate disassembly, the end of the bolt threaded portion is made spherical (168, *I*) or provided with a projection (Fig. 168, *II*) which allows the bolts to be removed by striking them on the end without damage to the thread.

Fig. 168, *III*, *IV* shows taper dowel bolts. The taper provides a very tight fit, which can be regained in each repeated assembly. Since these bolts do not perform the same fastening functions as cylindrical dowel bolts, they are used in combination with regular fasteners. A taper of 1/3 to 1/10 is used for this type of dowel bolts. Taper dowel bolts, even those seated very tight, are removed from the holes by striking light blows on the ends. As with cylindrical

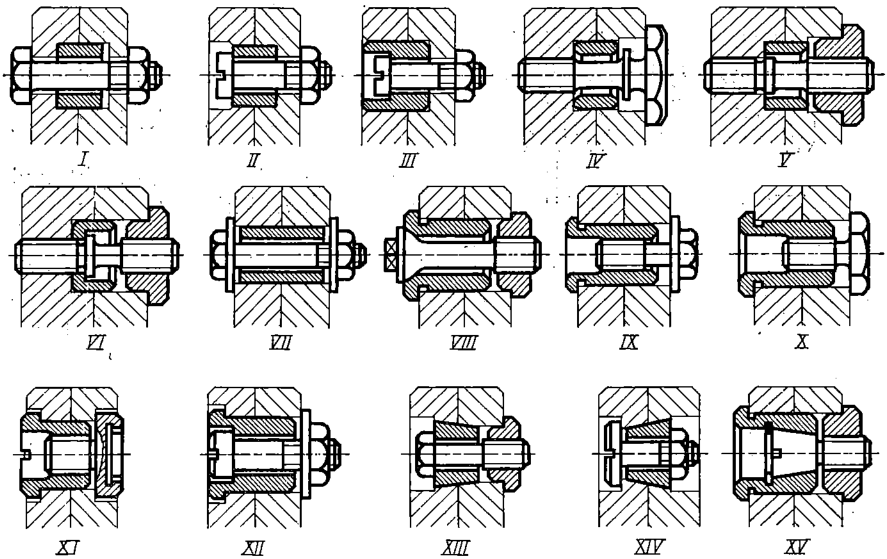


Fig. 170. Locating bushings

dowel bolts, the taper-dowel type has its threaded ends shaped so as to prevent damage to the thread.

Fig. 169, *I*, *II* gives examples of the use of dowel bolts for fastening several components.

Shown in Fig. 170 are assemblies with *locating bushings* used in conjunction with regular bolts and studs. Through holes for locating bushings should be preferred (Fig. 170, *VII-XII*). Fig. 170, *XIII*, *XIV* presents taper bushings. The design in Fig. 170, *XV* makes use of a bushing which has an internally tapered seat for a taper bolt; as the nut is tightened, the bolt expands the bushing, thereby providing its tight fit in the hole.

1.12. Differential-Thread Assemblies

In some special cases, use is made of bolts and nuts with a differential thread.

A differential screw (see Fig. 171) has two threaded portions. The threads of both portions are of the same hand (right or left), but of

different pitches (e.g. pitch s_1 of one thread is greater than pitch s_2 of the other). The first threaded portion is screwed into one of the assembled parts, and the second into the other. One revolution of the screw moves the parts towards each other by an amount equal to the difference between the pitches $s_1 - s_2$. The resultant travel of the parts will be $n\ (s_1 - s_2)$, where n is the number of screw revolutions required to fully tighten the joint. Therefore, a differential screw is equivalent to a plain screw with a very fine thread having a pitch of $s = s_1 - s_2$.

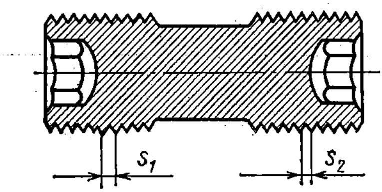


Fig. 171. Differential-thread bolt

Assemblies with differential screw threads have the following merits:

(1) The joint can be tightened hard with a limited torque;

(2) as the screw is loosened the assembled parts are forced to move apart (i.e. a differential screw serves as a natural remover);

(3) when turning the screw, very small displacements of the parts are obtained; therefore a differential screw allows fine positional adjustment of the assembled parts;

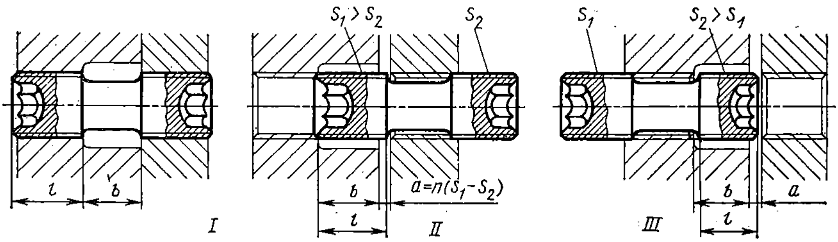


Fig. 172. Assembling with differential-thread bolt
I—after assembly; *II*—before assembly; *III*—before assembly (alternative assembly sequence)

(4) a bearing surface required to abut the screw head, which is necessary for regular cap screws, is not needed here; its function is performed by the threaded shank.

Differential thread makes it possible to obtain threaded assemblies with small radial dimensions, especially if driving elements (hexagons, splines, etc.) are arranged as sockets in the screw. Assembling with differential screws has its special features which must be taken into account.

Differential screws are used singly since the tightening of several such screws placed in parallel is extremely difficult and may result in jamming of the assembled parts and the screws. Prior to assembly, the parts should be positioned relative to each other at a distance equal to $n\ (s_1 - s_2)$.

Fig. 172, *I* illustrates the fastening of two components with a differential screw which has threaded portions of the same diameter. The initial position of the screw and the components is shown in Fig. 172, *II*. The screw is first turned into the component which has the finer thread with pitch s_2 until the threaded portion of the screw has nearly left its tapped hole (only one or two threads remain in the hole). The second component is then brought to the first one so that its threaded hole comes in contact with the second threaded portion of the screw; distance a between the two components should in this position be equal to $n(s_1 - s_2)$. A recess should be provided in the second component to accommodate the respective threaded end of the screw, the depth of the recess b being somewhat greater than $l - a$, where l is the length of the screw threaded portion.

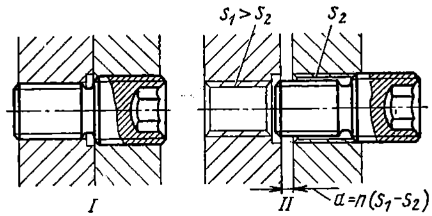


Fig. 173. Assembly with bolt having threads different in pitch and diameter
I—after assembly; *II*—before assembly

If s_2 is greater than s_1 (Fig. 172, *III*), the screw is first turned into the component with the recess.

The assembly procedure is simplified by providing threads of different diameter (Fig. 173) so that the minor diameter of the larger

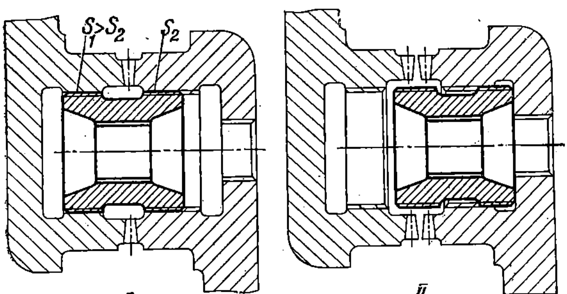


Fig. 174. Connecting crankshaft parts with differential-thread nut
I—after assembly; *II*—before assembly

thread is greater than the major diameter of the smaller thread. That obviates the need for the recess and for the initial running of the screw through the associated thread.

Fig. 174, *I* shows an assembly where two parts of a crankshaft separable at the main journal are joined by a differential-thread insert and coupled by means of triangular face claws. A limitation

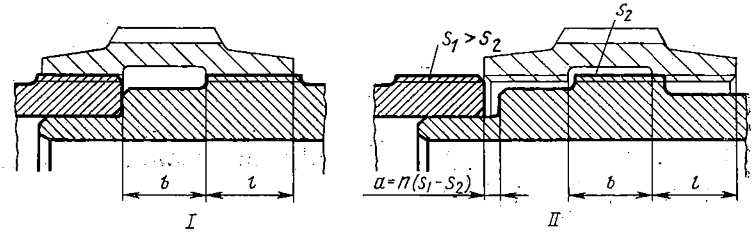


Fig. 175. Connecting tubular parts with differential-thread nut
I—after assembly; *II*—before assembly

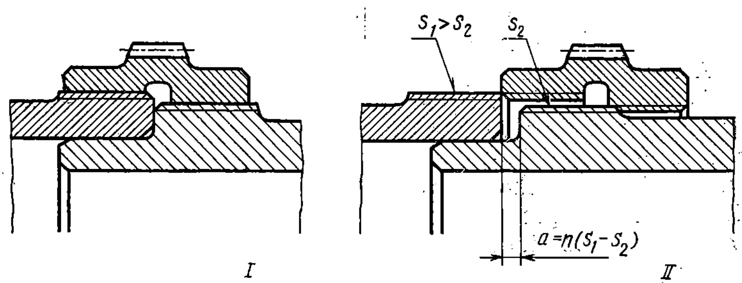


Fig. 176. Connecting tubular parts with nut having threads different in pitch and diameter
I—after assembly; *II*—before assembly

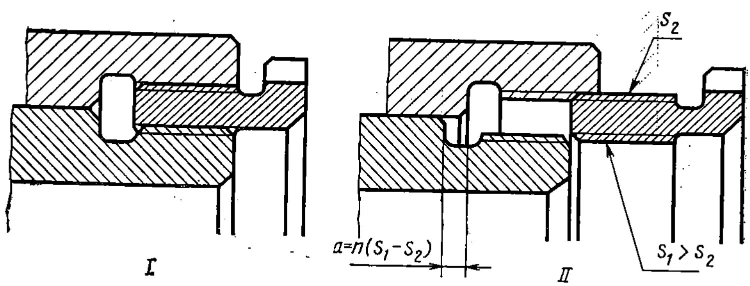


Fig. 177. Hub fixed on shaft with differential-thread nut

on the axial size of the assembly has necessitated the positioning of the insert inside the crankshaft.

The components as arranged before assembly are shown in Fig. 174, *II*. The insert is screwed fully into the right part of the crankshaft. Then, the insert is turned by its internal splines into the thread of the left part, and so the two parts are brought together. If the threads in both parts are equal in diameter, recesses are pro-

vided therein to admit one of the insert's threaded portions prior to assembly.

Connections of two tubular parts by means of a nut with differential threads of the same and different diameters are shown in Figs. 175 and 176, respectively; the necessary positional relationships to be held during assembly are given there.

Fig. 177 illustrates the fixing of a hub on a shaft by a differential nut. The arrangement is often used where the hub has to be forced from the shaft in dismantling.

1.13. Assemblies with Opposite-Hand Threads

Fasteners with opposite-hand threads (right-hand and left-hand) are used relatively seldom, e.g. when the assembled parts must be quickly moved towards and away from each other or when they should be positionally adjusted relative to each other.

One revolution of the fastener produces a displacement of the assembled parts equal to twice the pitch of the threads. The tighten-

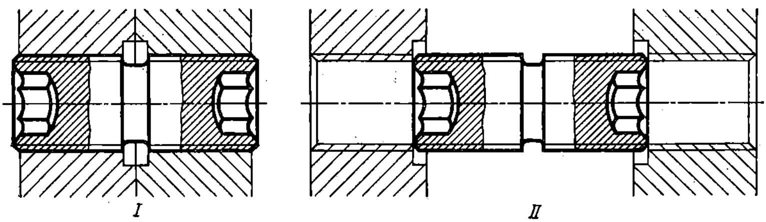


Fig. 178. Assembling with bolt having threads of opposite hands
I—after assembly; *II*—before assembly

ing force drops in the same proportion (at a given torque). For this reason, fasteners with opposite-hand threads are hardly ever used for high-force tightening.

Assemblies with opposite-hand threads are shown in Figs. 178-180.

The most extensive application of opposite-hand threads is in *turnbuckles*, which are used for adjusting the tension of cables and connecting rods, for adjusting relative axial position of parts, etc.

Fig. 181 presents general-purpose turnbuckles. The one shown in Fig. 181, *I* is a cylindrical hollow sleeve which is turned with a piece of rod passed through the cross holes. In another variation (Fig. 181, *II*), the turnbuckle is shaped for uniform strength along its length and provided with two pairs of cross holes at right angles to one another. The turnbuckle in Fig. 181, *III* is made from a hexagonal bar, whereas that in Fig. 181, *IV* has a narrow hexagonal portion. The most widely used version (Fig. 181, *V*) is made from a forging with an elongate opening for turning. The turnbuckle

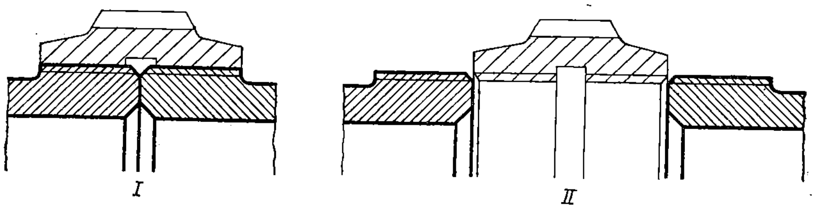


Fig. 179. Connecting tubular parts with nut having threads of different hands
I—after assembly; *II*—before assembly

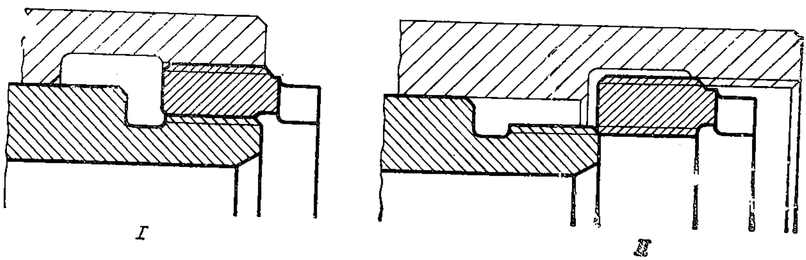


Fig. 180. Hub fixed on shaft with nut having internal and external threads of opposite hands
I—after assembly; *II*—before assembly

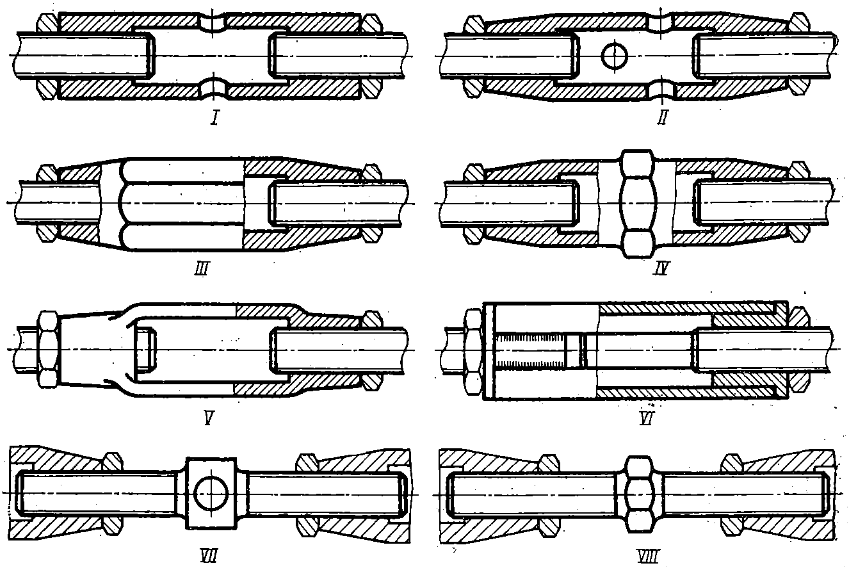


Fig. 181. Turnbuckles

shown in Fig. 181, *VI* is a weldment comprising two bushings connected with two preformed arcuous strips.

Reverse-type constructions are shown in Fig. 181, *VII*, *VIII*. Here, the turnbuckle is a shank with a right-hand and a left-hand thread on the ends, which are screwed into respective parts to be assembled; the shank is provided with holes or a hexagonal collar for a spanner. Turnbuckles of this type are used rarely, because their adjustment range is smaller (by the extent of the turning feature) than that of the foregoing designs of the same size.

1.14. Bearing Surfaces under Nuts and Bolt Heads. Wrench Clearances

Given in Fig. 182 are dimensional relationships between bearing surfaces and associated nuts (or bolt heads) placed in counterbores (Fig. 182, *I*), on bosses (Fig. 182, *II*), and flanges (Fig. 182, *III*), these surfaces being counterbored, milled or turned, respectively.

The numerical values obtained from these relationships should be rounded off to the nearest greater whole number. In the case of counterbores (Fig. 182, *I*), the value of D_1 should be rounded off to the nearest greater diameter of a standard socket wrench.

Wrench Clearances. When designing threaded assemblies, an adequate space for spanner manipulation should be provided. This space should allow the spanner to be swung through no less than 90° for tightening and loosening the fastener.

Figs. 183-189 give the minimum distances between the centres of nuts (screw heads) and the nearest structural elements. These distances are determined from statistical analysis of dimensions of the most commonly used wrench types.

To ensure free manipulation with a spanner of any type, the following general rule can be recommended on the basis of the obtained relationships (Figs. 183-188): the minimum distance between the centre of the nut (or screw head) and the nearest wall must be equal to nut diameter D; when tightening with a closed end spanner in recesses (Fig. 186), the distance between the nut centre and the side walls must be $1.5D$.

In the case of a nut sunk in a deep counterbore (Fig. 189), the recommended minimum diameter of the counterbore can be $D' = 1.5\,D$ for a tubular spanner and $D' = 1.8D$ for a solid box spanner.

The foregoing relationships hold true where positional deviations of the relevant features can be neglected. Where the distances from the limiting walls to the surfaces from which the bolt holes are located in machining are long, deviations from the nominal position of the walls must be taken into account.

In castings, in the general case, a wall can be located from datum A, and the nearest bolt hole, from datum A', the latter being the machining location for drilling the hole (Fig. 190). Then, the actual minimum distance between the hole and the wall may be

$$a = (L_1 - \Delta L_1) + (S - \Delta S) - (L_1 + \Delta L_2)$$

where L_1 = distance between hole centre and machining location; L_2 = distance between wall and its datum on casting; ΔL_1 = 'minus' tolerance on dimension L_1; ΔL_2 = 'plus' tolerance on dimension L_2;

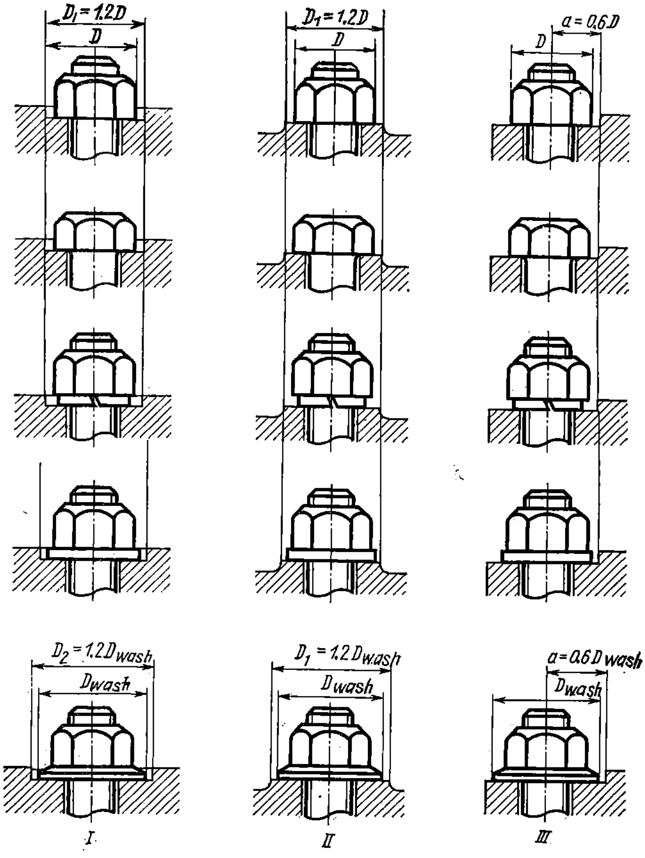


Fig. 182. Dimensions of bearing surfaces under nuts and bolt heads

S = distance between datum A and datum A' (machining location); ΔS = 'minus' tolerance on dimension S.

Expressed in a different way,

$$a = (L_1 + S - L_2) - (\Delta L_1 + \Delta L_2 + \Delta S)$$

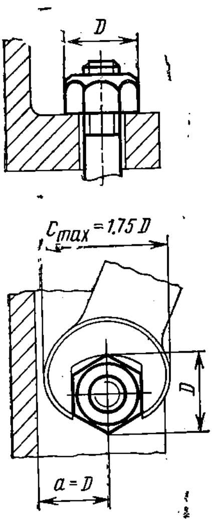


Fig. 183. Tightening with open-ended spanner swung through 180°

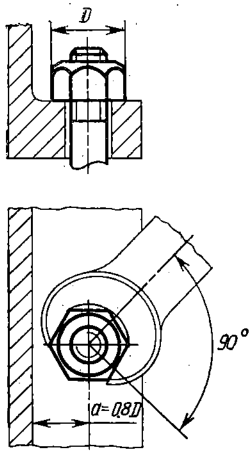


Fig. 184. Tightening with open-ended spanner swung through 90°

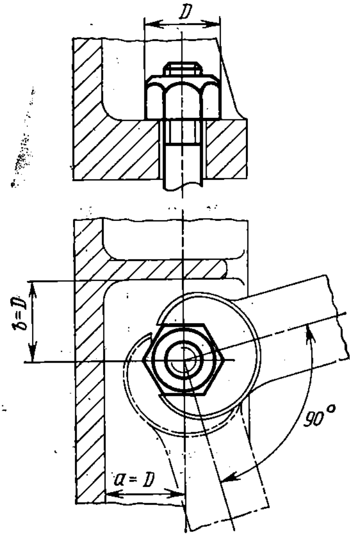


Fig. 185. Tightening with open-ended spanner swung through 90° in a corner

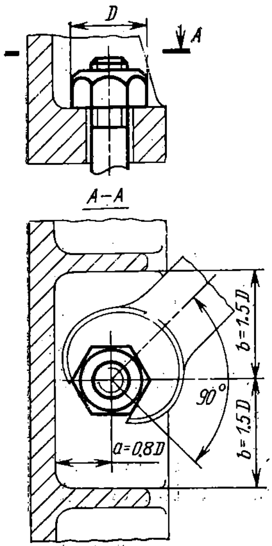


Fig. 186. Tightening with open-ended spanner swung through 90° in a recess

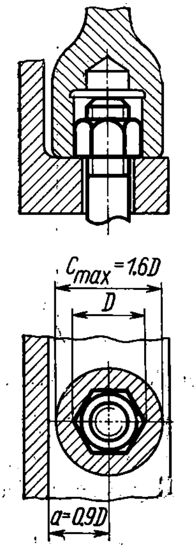


Fig. 187. Tightening with solid box spanner

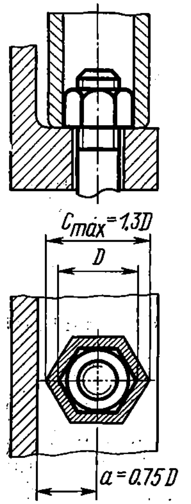


Fig. 188. Tightening with tubular spanner

Dimension a should have a definite minimum value to ensure free manipulation with a spanner. Let us assume that $a_{\min} = D$, where D is the width across corners of the nut (or screw head).

Then the specified nominal value

$$a_{nom} = L_1 + S - L_2$$

should be equal to

$$a_{nom} = D + \Delta L_1 + \Delta L_2 + \Delta S$$

Where the nut is placed into a deep seat with unmachined walls (Fig. 191), the actual minimum diameter of the seat, D', to be specified by the drawing should be

$$D' = D'_{\min} - \Delta D' - 2(\Delta L_h + \Delta L_s)$$

where $D'_{\min}$ = minimum diameter of seat determined by spanner size; $\Delta D'$ = absolute value of 'minus' tolerance on seat diameter;

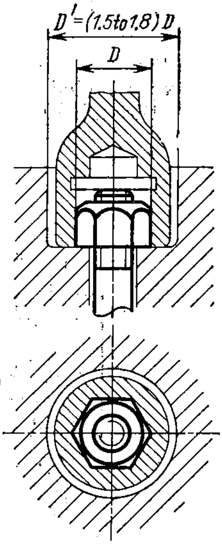


Fig. 189. Tightening a countersunk nut with solid box spanner

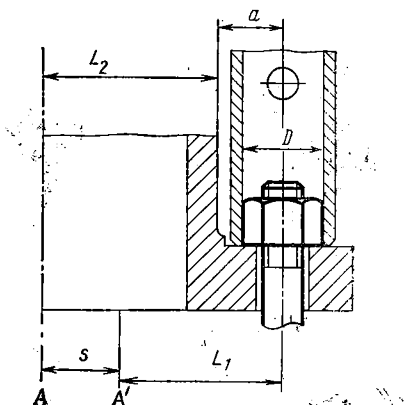


Fig. 190. Determining distance a

$\Delta L_h + \Delta L_s$ = maximum distance between nut and seat centres determined by tolerances on distance L from datum A to centres of bolt hole and seat.

If $D'_{\min} = (1.5\text{-}1.8)\,D$ as recommended above, the seat diameter will then be

$$D' = (1.5 \text{ to } 1.8)\,D + \Delta D' + 2(\Delta L_h + \Delta L_s)$$

The foregoing dimensional relationships hold for hand tightening. Modern methods of tightening with electric, pneumatic, and other

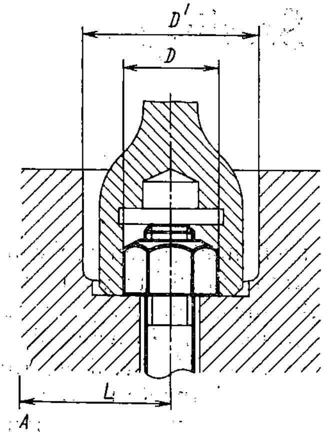


Fig. 191. Determining diameter D' of counterbore

types of nutrunners as a rule require greater wrench clearances. In units designed for mechanized assembly, wrench clearances should be compatible with the size of nutrunner heads.

Chapter 2

Sealing Movable Joints

Seals (packings) are used most extensively for packing tightening input and output shafts in machinery and equipment. They serve both seal-in and seal-out functions, i.e. prevent leakage of oil from a machine housing and protect the interior of the housing from the ingress of external agents, such as dust, dirt, and moisture. The latter function is particularly important for equipment operating outdoors and exposed to aggressive environments.

Seals are of especial importance in machines and apparatus having containers with chemically active substances (in the chemical engineering industry), food stuffs (in the food industry), etc. Dependable sealing of these containers is vital for the appropriate operation of the machinery.

Another application of seals is the tightening of containers with gases or liquids under high pressure or vacuum. In rotor-type machines (steam and gas turbines, radial- and axial-flow compressors, etc.), it is essential to seal the rotating shafts and rotors and in piston-type machines, it is essential to seal the reciprocally moving parts (pistons, plungers, rods, etc.).

In seeking adequate solutions, design engineers have developed a wide variety of sealing units and systems. We shall describe here typical constructions of seals used in general engineering. These constructions, which also form a base for special-purpose seals, may be classified into two groups: contact-type (or sliding-contact) seals and contactless seals.

The first group comprises designs where sealing is provided by direct contact between the movable and stationary members of a sealing unit. Related to this group are felt-ring (or groove) packings, stuffing boxes, lip-type packings, split metal rings, axial seals, etc.

Seals of the second group have no contact between their members. The sealing effect is achieved by means of centrifugal forces, hydrodynamic phenomena, etc. This group includes labyrinth seals, baffle-type elements (thread or plates), oil traps of different types, etc.

Sliding-contact seals provide higher tightness of joints packed than contactless seals. Their shortcomings (rather low permissible speeds for relative motion of their members, fast wear, and, conse-

quently, loss of sealing capacity) are eliminated by the adjustment of contact pressures between interactive members, by the appropriate selection of material for frictional surfaces, and by compensation for wear through the use of flexible elements.

Reference books frequently give the values of allowable working speeds for different types of seals. This practice seems to be questionable. Safe speeds are determined by many factors (the properties of the liquid being packed, the lubricating conditions, the amount of contact pressure, the material of rubbing surfaces, the proper assembly, etc.), and an optimum combination of these factors will substantially extend the range of speeds for reliable operation of a seal.

Contactless seals have no limitation on the speeds of relative motion; their service life is unrestricted; the sealing capacity is generally lower than that of sliding-contact seals; complete tightness can only be achieved by the use of additional sealing means.

2.1. Sliding Contact Seals

2.1.1. Groove Seals and Stuffing Boxes

Groove seals and stuffing boxes are obsolescent types of seals. Their main disadvantages consist in fast wear of packing that results in lowered sealing capacity, and also in unsuitability for operation at high peripheral speeds. They have been in use, however, in noncritical machine units because of simple design and low cost. A groove-type shaft seal is a circular recess around the shaft, which is filled with a packing material. The groove-type seal provided with a gland follower designed to compress the packing material is called the stuffing box. Used as packing materials are cotton yarn, oil-impregnated cord, felt, asbestos, and the like materials with additives of metal powders (lead, babbit), graphite, molybdenum disulphite, and other self-lubricating substances.

Fig. 192 shows the simplest groove-type shaft seals mounted directly in bearing housings (Fig. 192, *I-IV*) or in intermediate parts (Fig. 192, *V-VIII*).

A simple groove packing, i.e. seal in which a ring of packing material is let into a trapezoidal-section groove with a standard profile angle of 15° ± 1° of arc, is illustrated in Fig. 193, *I*. The groove is shaped to the trapezoidal form in order that the cylindrical packing ring, made of felt, might become compressed by its proper elastic forces towards the centre, pushed into the groove, and thus might tightly embrace the shaft.

The packing operates in contact with the shaft or with an intermediate sleeve; for increased dependability and service life, the surface of the shaft (or sleeve) should have a hardness of no less

than 45 Rc and a roughness not over 0.32 to 0.65 μm Ra. A reverse arrangement in which the packing ring functions in contact with

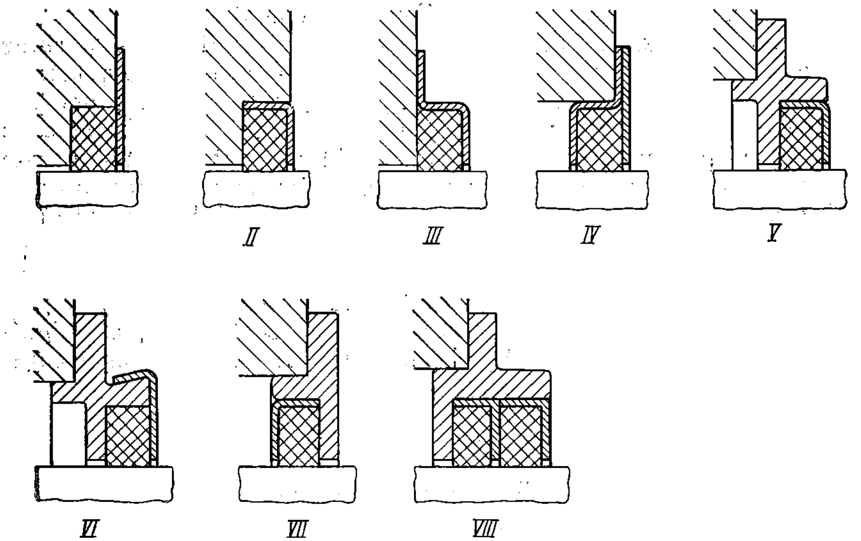


Fig. 192. Cylindrical-groove packings

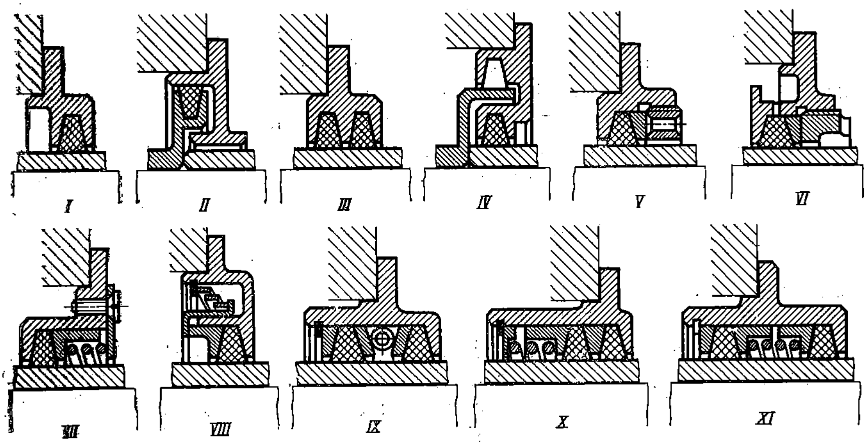


Fig. 193. Conical-groove packings

the housing (Fig. 193, *II*) is used rarely because here the sliding speed is higher than in the convectional groove packing.

To improve the reliability of sealing, use is made of two packing rings disposed one by one (Fig. 193, *III*) or, where the space is

limited axially, one over the other (Fig. 193, *IV*). The packing is compressed with a gland follower to make up for wear occurring in service (Fig. 193, *V*, *VI*).

The life and reliability of a stuffing box is substantially improved if the packing is lubricated, even slightly, because the lubrication decreases the coefficient of friction and generation of heat and makes

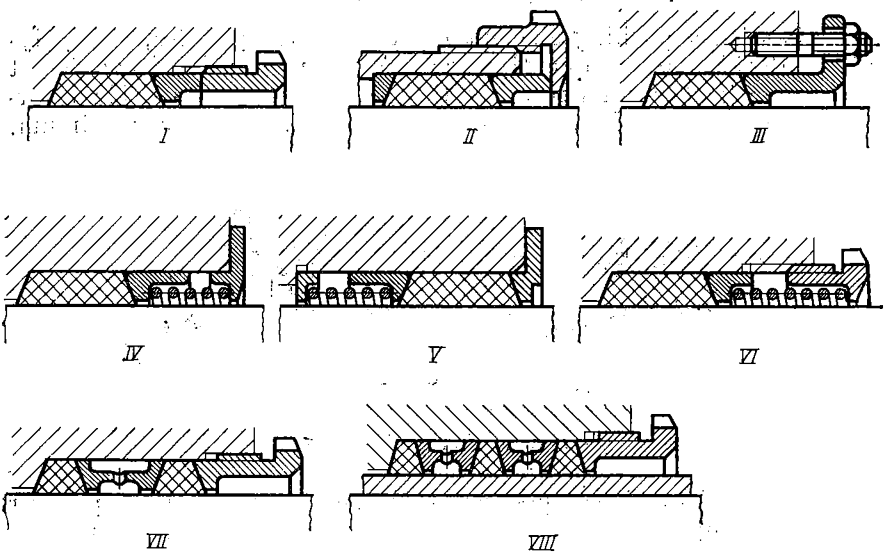


Fig. 194. Stuffing boxes for operation at high pressures (liquid-drain passages in seals of views *VII* and *VIII* are not shown)

for better tightness. The construction shown in Fig. 193, *VI* features radial passages in the stuffing box, through which oil is supplied from the chamber being sealed.

Stuffing boxes with periodically tightened gland followers are objectionable because they call for continual attention of service personnel. Moreover, owing to the lack of proper skill in handling such a type of sealing unit, it can be tightened in excess of what is necessary, which will lead to its overheating and failure.

Stuffing boxes of more advanced design feature the gland followers tightened automatically by spring pressure (Fig. 193, *VII* and *VIII*).

Fig. 193, *IX-XI* displays spring-tightened double-packing stuffing boxes.

To seal against liquids, steam, and gases at high pressures, use is made of stuffing boxes with an increased length of packing compressed by means of a sleeve nut (Fig. 194, *I*), a union nut (Fig. 194, *II*), a flanged sleeve (Fig. 194, *III*), or a spring (Fig. 194, *IV-VI*).

Double- or multi-packing stuffing boxes (Fig. 194, *VII* and *VIII*, respectively) provided with intermediate spacer sleeves between the packing rings and with passages for the drainage of liquid working its way through the packings that face the chamber being sealed are applicable where leakage through the seal is to be totally excluded.

2.1.2. Hydrocarbon-Plastic Seals

Seals with a packing element made of thermoplastics, e.g. polyvinylchloride, are frequently used. A thermoplastic sleeve is let into a confined circular recess in the housing (Fig. 195, *I*). The clearance between the shaft and the sleeve hole is held to a minimum.

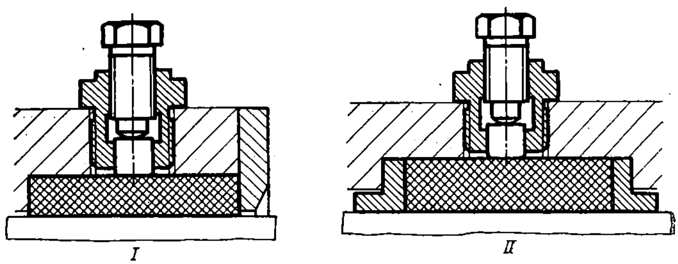


Fig. 195. Thermoplastic seals

The sealing sleeve is compressed on the shaft with a lapped plunger acted upon by a screw; the pressure is imparted by the plunger to the whole mass of the thermoplastic, thus causing the sleeve to tightly embrace the shaft.

To prevent the thermoplastic from extrusion into the clearance between the shaft and the housing, rings made from an antifriction metal are placed in the face portions of the housing's annular groove (Fig. 195, *II*). Sliding fit is provided between the shaft and the rings, and the latter have a certain freedom to move radially in the housing to prevent the sliding surfaces from wear due to the shaft's radial runout.

2.1.3. Lip Packings

A lip packing is a ring made from a soft elastic material, provided with a sealing lip that encircles the shaft being sealed. Under the action of pressure in a chamber being sealed, the lip tightly embraces the shaft with a force proportional to the pressure (Fig. 196, *I*). To secure stable tightness, the lip is compressed on the shaft with a garter spring (not shown in Fig. 196).

A lip packing should present its lip to the fluid pressure; in the reverse disposition (Fig. 196, *II*) the pressure will force the lip

away from the shaft. Where double-sided sealing is required, two packings are installed so that their lips are opposite to each other

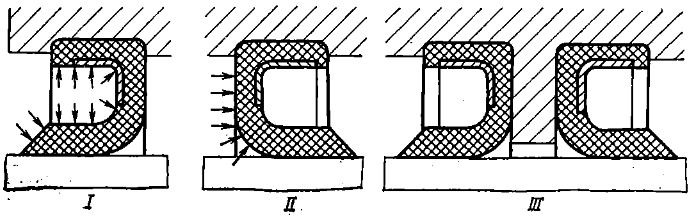


Fig. 196. Principle of lip-type packing

(Fig. 196, *III*). The packing's outside portion is tightly secured to the housing.

For some applications a packing is made with two lips, one sealing the shaft and the other, the packing proper against the housing (Fig. 197).

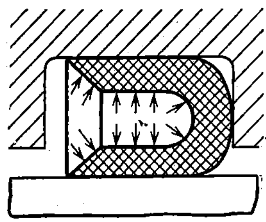


Fig. 197. U-packing with two sealing lips

Various shapes of lip packings are shown in Fig. 198, *I-XII*.

Lip packings were formerly produced from the best kinds of cup leather, which was subjected to steam softening and shaping to the required form. Fig. 199 presents different types of seals with leather packings.

Fig. 200 exemplifies the use of lip packings in axial seals.

Fig. 201 shows tandem arrangements of lip packings in seals operating against high-pressure liquids, steam, and gases.

Lip packings are now produced most often from polyvinylchloride or fluoroplastics, which outperform leather in elasticity and wear

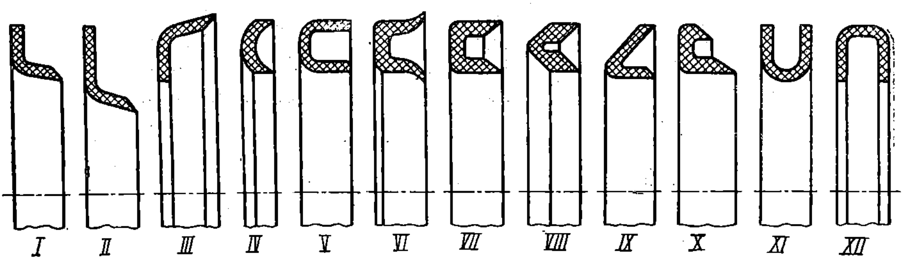


Fig. 198. Shapes of lip-type packings

resistance. Polyvinylchloride packings withstand temperatures up to 80°C, and fluoroplastic packings, up to 300°C.

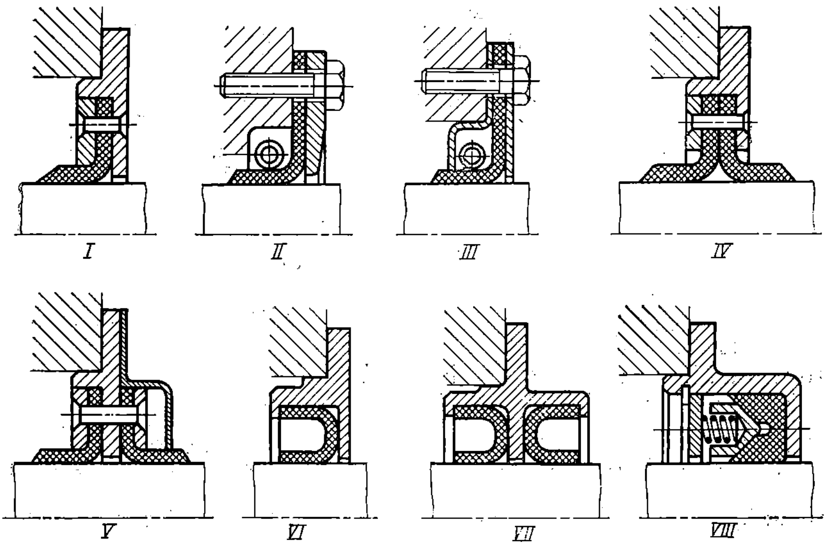


Fig. 199. Varieties of seals with leather lip-type packings

I, *VI*—single-sided; *II*, *III*—single-sided with lip compressed by garter spring; *IV*, *V*, *VII*—double-sided; *VIII*—single-sided with spring pressed lips

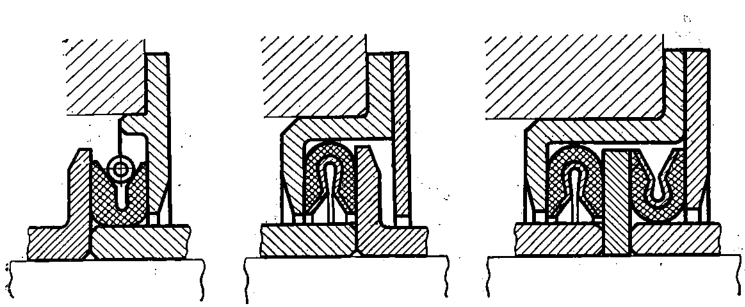


Fig. 200. Lip packings in axial seals

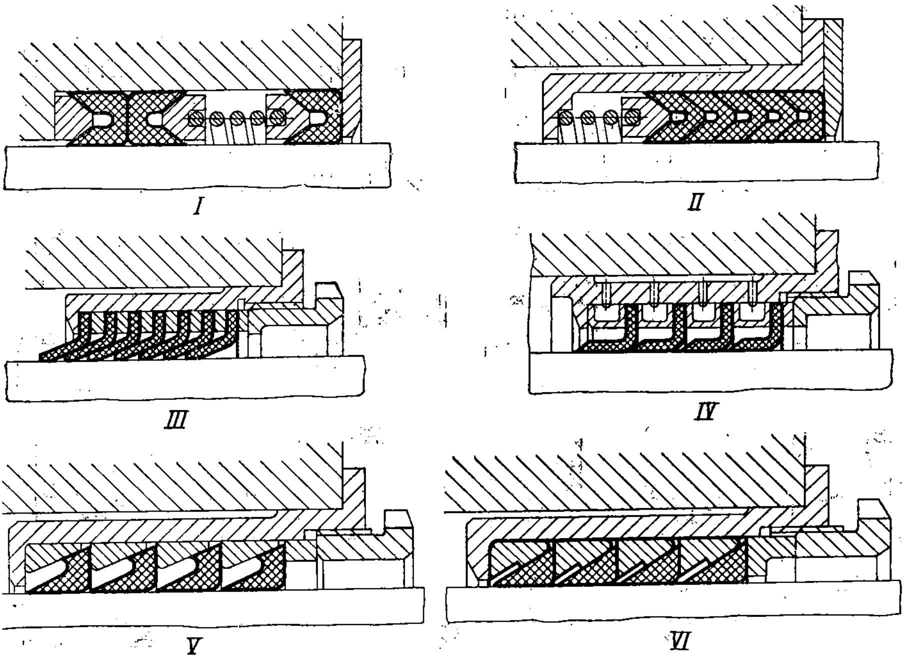


Fig. 201. Tandem arrangement of lip packings in high-pressure seals
I, II—with spring-pressed V packings; *III*—with spacer washers; *IV*—with spacer rings and means for draining fluid from intermediate spaces (drain holes are not shown); *V, VI*—with conical tightening rings

2.1.4. Rotary-Shaft Oil Seal Units

This type of shaft seal which is extensively used in the engineering industry is a separate lip-packing unit mounted as a whole into a housing; the packing is produced from appropriate synthetic materials. The sealing lip is compressed around the shaft by an annular helical spring (garter spring) at a specified pressure.

Fig. 202 shows various types of oil seal units (the first ones are the earlier designs).

The units in Fig. 202, *I-VIII* with the packing encased in a sheet steel frame (which may be very hard to assemble) are now almost out of use. Their chief disadvantage is the difficulty of securing tightness of the unit fitted into the housing; this difficulty is due to a low dimensional accuracy obtainable by stamping the unit's frame. For this reason, such units need to be additionally tightened with sealing greases.

In modern designs, the locating portion of the unit forms an integral part of the packing proper (Fig. 202, *IX* and the others). Here,

tightness of the unit in the housing seat is achieved easily even with considerable variations in the size of the locating diameter owing to the packing-material elasticity. The required radial stiffness is obtained by embedding a steel annular frame into the packing body.

The packing rings are made with a single sealing lip (Figs. 202, *X* and 203, *I*, *II*), two lips (Fig. 202, *XI*, *XII*), and with several lips (Fig. 202, *XIII*). In the design shown in Fig. 202, *XIII*, a garter

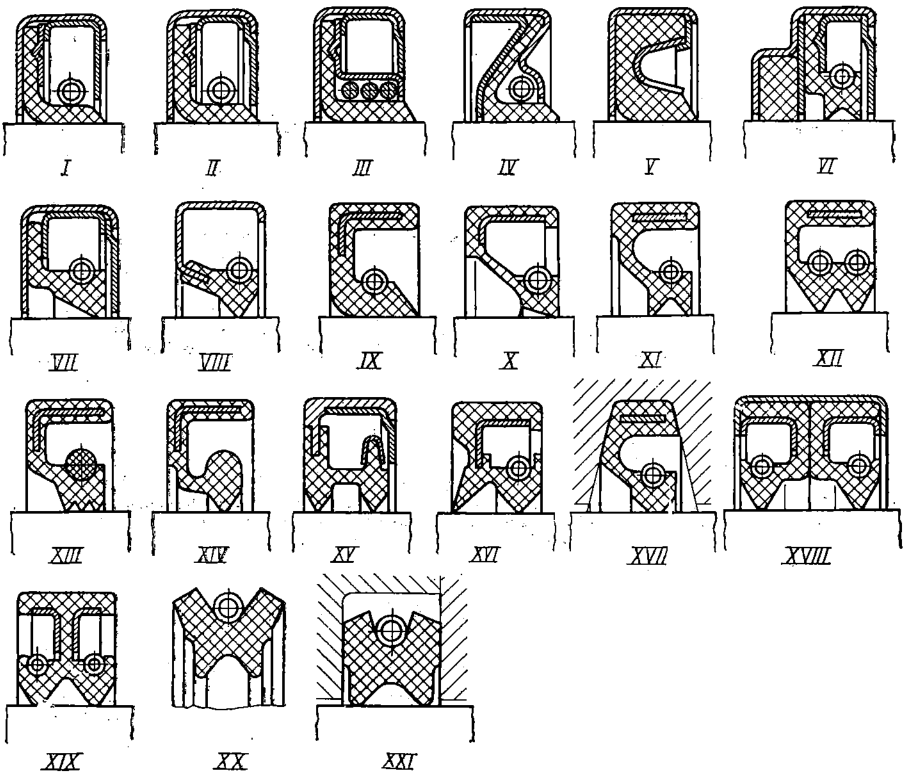


Fig. 202. Rotary-shaft oil seal units

spring is replaced by a ring of an elastic synthetic material. The required elasticity can be imparted to the packing by a projection of a semi-circular or elongated cross-section provided around the lip (Fig. 202, *XIV* and *XV*, respectively); for higher stiffness, the projection in the second design is enclosed in a stamped frame.

Fig. 202, *XVI* presents a sound design of a seal unit with two lips; one compressed by a garter spring, seals the shaft, and the other prevents the entry of dirt into the unit from outside. Fig. 202, *XVII*

shows a seal unit inserted radially into a conical groove in the housing, and Fig. 202, *XVIII*, *XIX* shows double seal units. An ingenious design of a double-lip seal unit is illustrated in Fig. 202, *XX*, *XXI*. In the free state, the packing has the shape shown in Fig. 202, *XX*. When the unit is mounted into the housing (Fig. 202, *XXI*), the sealing lips spread apart, producing tightness on the shaft surface; the tightness is maintained by a garter spring.

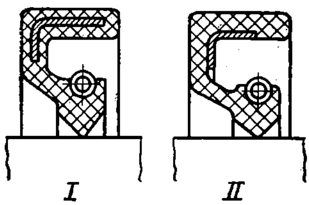


Fig. 203. Seal-unit design (GOST 8752-70)

Oil seal units are made by press- or injection-moulding from elastic, wear-resistant, and oil- and chemically-resistant plastics and rubber, with the inner metallic insert placed in the mould. Garter springs are produced from spring wire of 0.2 to 0.5 mm in diameter and subjected to hardening and medium tempering; they may be protected by cadmium or zinc plating, or else be made from bronze.

Methods for connection of the garter-spring ends are illustrated in Fig. 204. In the design shown in Fig. 204, *I*, one end of the spring is wound so as to form a step. The connection is made by screwing the stepped end, previously twisted in the direction opposite to the hand of its coiling, into the coils of the other end.

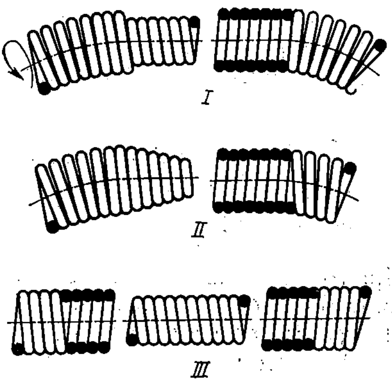


Fig. 204. Methods for connecting garter-spring ends

A spring may be produced with a tapered end, which will simplify its screwing-in (Fig. 204, *II*); the connection of the ends may also be effected with the aid of a separate coiled insert (Fig. 204, *III*).

Methods for mounting seal units into their housings are illustrated in Fig. 205. The seal unit and the housing are joined, according to Fig. 205, *I*, through elastic radial compression of the unit as it is fitted in place; such a conjunction, however, proves unreliable. In the design according to Fig. 205, *II*, the seal unit, previously compressed, is let into a suitable circular groove in the housing; the height of the shoulder on the entering side of the groove should not exceed the permissible amount of the seal unit's elastic deformation.

Fig. 205, *III* shows a better mounting method: the seal unit is fixed axially with a washer-type cover screw-fastened to the housing.

To prevent the unit from turning in the housing and secure tightness, the seal unit is mounted so as to be compressed axially by about

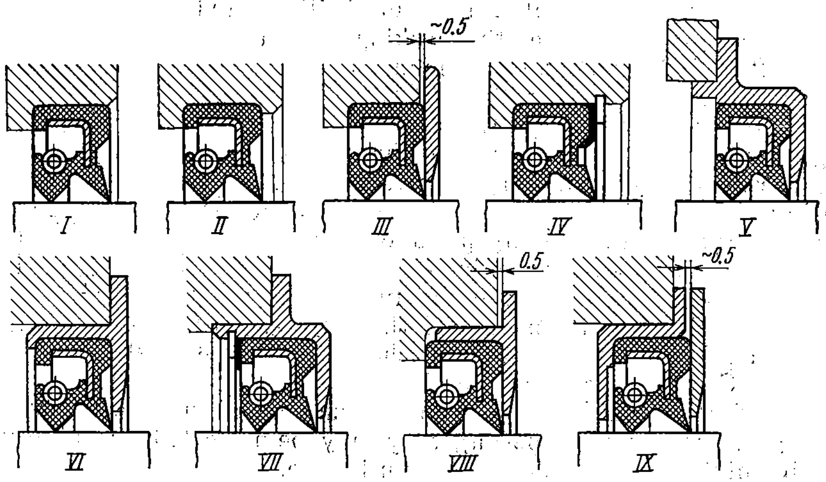


Fig. 205. Oil seal-unit mountings

0.5 mm. Fig. 205, *IV* displays a similar installation, where the seal unit is locked in place with a suitably shaped washer and a snap ring. Seal units mounted in intermediate casings are shown in Fig. 205, *V-IX*.

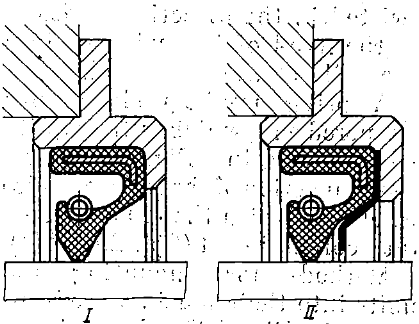


Fig. 206. Seal units mounted for operation against high pressures

I—incorrect; *II*—correct (packing is prevented from turning inside out by a bearing disc)

In the applications where seal units are to be exposed to high pressures, proper care should be taken not to allow the sealing lips to turn inside out. To this end, the use of a bearing disc shaped in accordance with the configuration of the packing ring is advisable (Fig. 206). The part surfaces to work in contact with the sealing lips should have a hardness not lower than 45 Rc and a roughness not above 0.16 to 0.32 µm Ra.

Fig. 207, *I-III* shows instances of seal-unit installation. In the second and the third instances, oil must not be allowed to work its way through the clearance between the shaft and the sleeve (or the hub of the part mounted on the shaft). For this purpose, the sleeve (or hub) end face *a* must be machined in a strict squareness with the

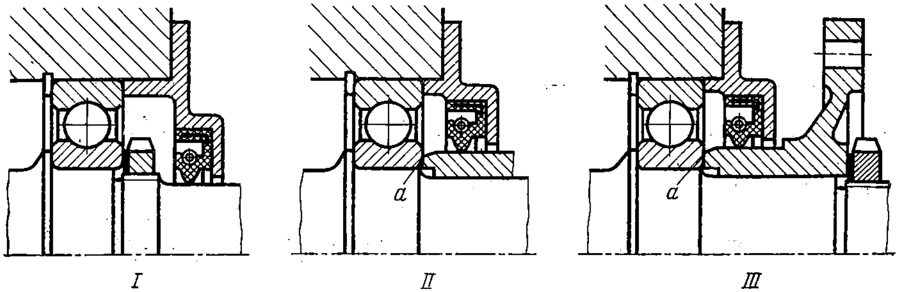


Fig. 207. Instances of seal-unit installation
I—on shaft; *II*—on intermediate sleeve; *III*—on hub of a part carried by shaft

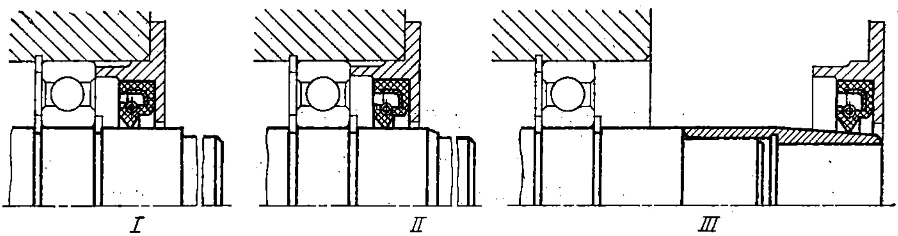


Fig. 208. Fitting oil seal units in place
I—difficult; *II*—eased by an entering chamfer; *III*—seal-unit mounting means for stepped shaft

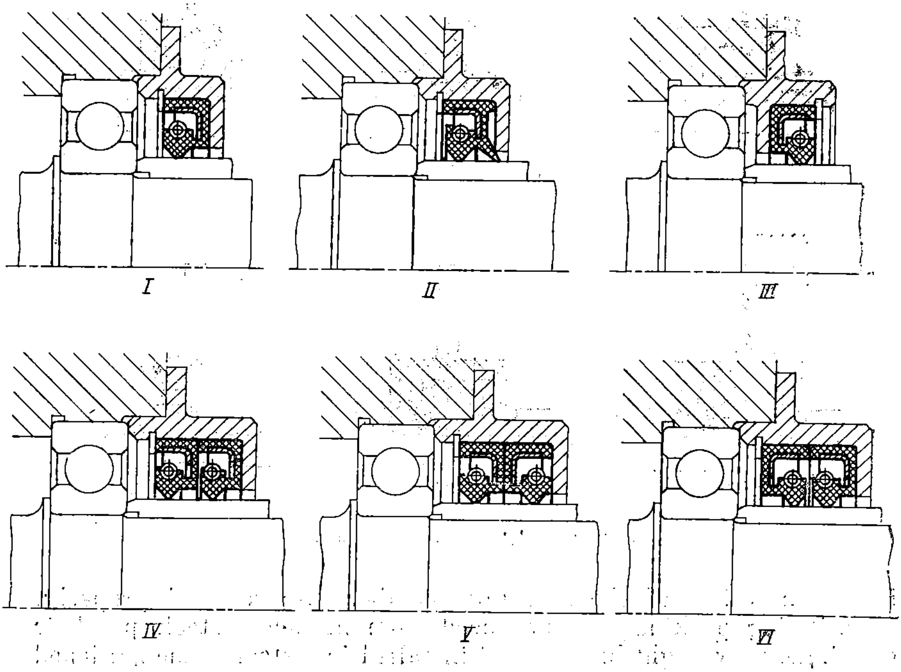


Fig. 209. Oil seal units in antifriction bearing assemblies
I, *II*, *IV*—for housing under pressure; *III*—for housing under vacuum; *V*, *VI*—double arrangement

hole axis, and its surface roughness must not exceed 0.63 to 1.25 μm Ra. Complete tightness will be achieved by coating the faces with sealing greases or by placing sealing gaskets.

For simpler assembly, the shafts that receive a seal unit should be provided with slightly sloping entering chamfers (Fig. 208, *II*). This will make unnecessary the use of special mounting devices, e.g. a mounting sleeve (Fig. 208, *III*).

Where a seal unit is to operate on an intermediate sleeve or on the hub of a part carried by the shaft (see Fig. 207, *II*, *III*), the entering chamfer on the sleeve or the hub is essential because here the assembly method shown in Fig. 208, *III* cannot be practised.

Fig. 209 illustrates oil seal units in ball bearing assemblies.

2.1.5. Split Metal-Ring Packings

Split metal-ring packings (Fig. 210) are fairly reliable; they can withstand high pressure differentials and have a long service life

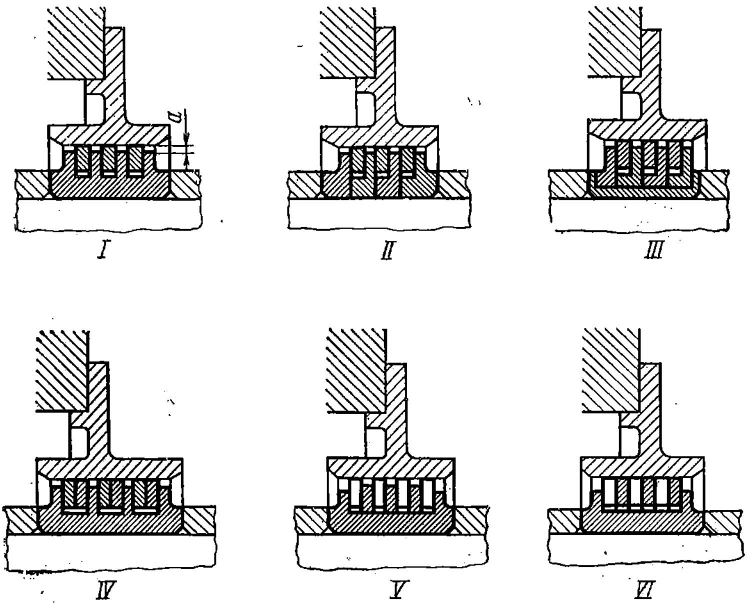


Fig. 210. Split metal-ring packings

if the materials of the seal-unit members are selected properly.

Split-ring packings are produced from hardened steel, pearlitic cast iron, or wrought bronze, and installed in a steel frame hardened to 40-45 Rc. The outer sleeve is made from a case-hardened (carbu-

rized or nitrided) steel. The packing rings are fitted into the frame grooves with an axial clearance of 0.005 to 0.020 mm. The clearance a between the outer surface of the frame and the sleeve hole (Fig. 210, *I*) is 0.5 to 1.0 mm.

The packing rings are installed into the outer sleeve by a slight interference fit. In operation, the packings are immovable or slightly turn in the sleeve. The difference of pressures on the packing rings

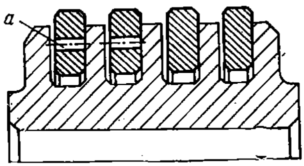


Fig. 211. Discharge holes in split rings

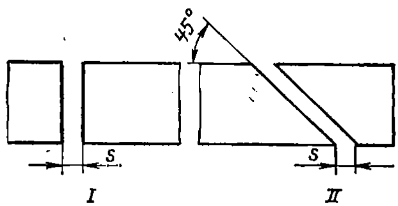


Fig. 212. Types of joints in split rings

keeps their faces close against the frame-groove walls. Two or three packing rings are normally used; at higher pressure differentials, the number of packing rings is increased to five-six.

In multi-ring packings operating under high pressure differentials, the first ring, next to the chamber being packed, is loaded most; during service, a stepped recess forms in the ring face as it bears against the groove wall.

Discharge holes a (Fig. 211) are made in the first packing rings (and sometimes in several rings consecutively, beginning from the first); in this way, load is spread uniformly among the whole set of rings, and oil is supplied to frictional surfaces (if oil-containing cavities are sealed).

The outside diameter d_0 of a split packing ring in the free condition is made so that the ring enters the sleeve with a slight interference: $d_0 = 1.02$ to $1.03d$, where d is the sleeve-hole diameter.

Split packing rings are commonly provided with straight joints (Fig. 212, *I*). In large-diameter rings, the joints are slanted at an angle of 45° (Fig. 212, *II*). The width of slit s in the free condition of the ring is so selected that it narrows in the ring installed in the sleeve to not less than 0.3-0.5 mm. With regard to the formula for d_0, we shall have

$$s = (0.3 \text{ to } 0.5) + \pi\,(0.02 \text{ to } 0.03)\, d \approx 0.5 + 0.08\, d$$

In the packings of this type operating at high temperatures, the slit width should be increased by an amount corresponding to the ring thermal elongation.

As is known from the theory of split metal rings, the b/d ratio (Fig. 213) should not exceed 0.05 if the rings are to be introduced into the grooves easily. With the rings made from high-grade hardened steels, this ratio may be increased up to 0.1. If $b/d > 0.1$, frames made up of a set of disc-type components are employed (Fig. 210, *II*). For convenient assembly, the discs with the installed packing rings are fixed on a sleeve of soft steel by flanging its ends (see

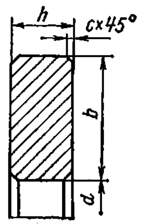


Fig. 213. Dimensioning a split ring

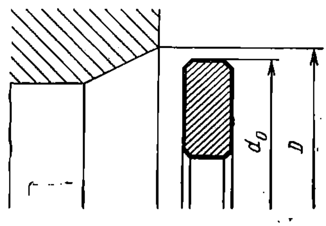


Fig. 214. Dimensioning the hole-edge chamfer for insertion of split rings

Fig. 210, *III*). The height-to-width ratio h/b in the split rings is usually equal to 0.5-0.7.

In some designs, the rings are installed in pairs (Fig. 210, *IV*) or in a row (Fig. 210, *V*, *VI*).

For easier driving of the packing rings into the outer sleeve, the sleeve-hole edge is provided with a slightly sloping chamfer. To avoid the use of special mounting devices, it is good practice to make the chamfer with a diameter D not smaller than the outside diameter d_0 of the packing ring in its free condition (Fig. 214).

2.1.6. Rubber Ring Packings

The application of rubber ring packings let into suitable grooves in the shaft or in an intermediate sleeve has been rather limited.

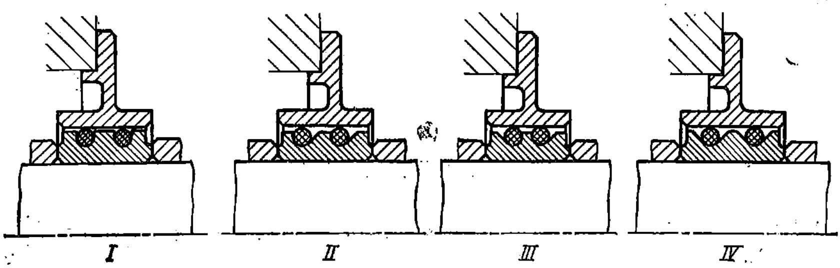


Fig. 215. Rubber ring packings

In the design presented in Fig. 215, *I*, the sealing effect is due to tightness between the outer surface of the packings and the flange-sleeve hole. The unit shown in Fig. 215, *II* operates similarly to lip packings. The packing rings are located in beveled grooves. Under the action of fluid pressure in the chamber being packed, the packing rings go up the beveled portions of the grooves to bear against the flange-sleeve surface. This packing is of a single-side type. To pack from both sides, the rings are mounted in grooves with oppositely directed beveled portions (Fig. 215, *III*) or double-beveled grooves (Fig. 215, *IV*).

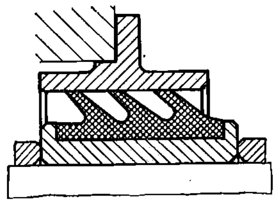


Fig. 216. Centrifugal-action rubber ring packing

Fig. 216 shows a packing unit that employs the centrifugal effect. A rubber ring is provided with several slanting lips, which are brought up against the sleeve by the action of centrifugal force; the built-up pressure increases as the square of rotational speed. The ring packings are made of soft grades of oil- and heat-resistant synthetic rubber.

Owing to fast wear and unstable tightening forces, rubber ring packings are rather unreliable in service.

Rubber rings are used rather often as reciprocating-rod packings.

2.2. Contactless Seals

Gap Seal. An annular gap between the shaft and its housing seat is the simplest type of contactless seal (Fig. 217). The packing effect of the gap is in direct proportion to its length and in inverse pro-

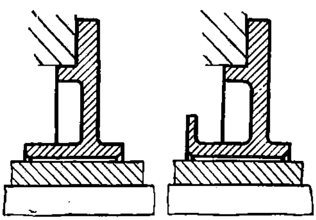


Fig. 217. Gap seals

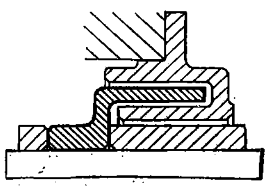
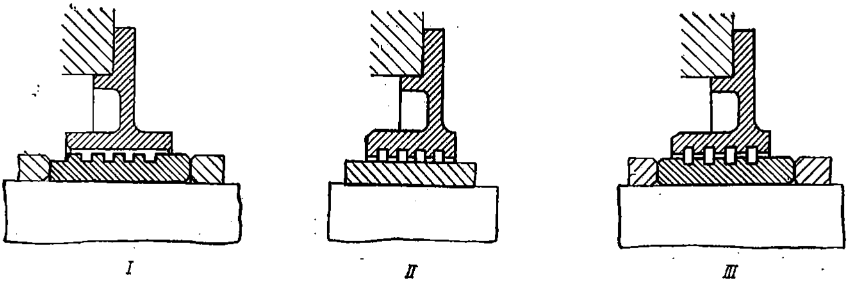


Fig. 218. Double-level gap seal

portion to the amount of the radial clearance. In practice, the gap with the dimensions that render the seal effective is difficult to obtain. To increase the gap length in a confined space, a double-level seal is sometimes used (Fig. 218).

The effectiveness of the gap seal is improved with the use of circular grooves made in the rotary shaft (Fig. 219, *I*), in the stationary

sleeve (Fig. 219, *II*), or in both (Fig. 219, *III*). This type of seal is often called *labyrinth packing*, although its principle has, actually, nothing in common with that of labyrinth packing. Here, the pur-



g. 219. Labyrinth-type gap seals

pose of the grooves is to form projections on the shaft that will throw the oil into the gap between the shaft and the sleeve by the action of centrifugal force. If the grooves are made in the stationary sleeve, the discharge of oil has to be provided for at the bottom.

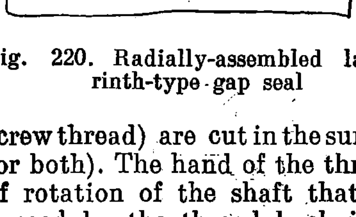


Fig. 220. Radially-assembled labyrinth-type gap seal

Fig. 220 shows an example of gap seal wherein the shaft projections are interlocked with the grooves of the sleeve. The seal allows assembly in a radial direction only.

Baffle-thread seals (Fig. 221, *I-III*) are used for packing liquid-containing chambers. Helical grooves (usually, multistart screw thread) are cut in the surfaces of the shaft or the stationary sleeve (or both). The hand of the thread must be so related to the direction of rotation of the shaft that the liquid being packed (e.g. oil) is turned by the thread back into the housing. The packing is non-reversible; if the direction of rotation is changed, the thread will drive the oil out of the housing.

Fig. 222 presents a multi-level baffle-thread packing. Thread profiles used in these packings are shown in Fig. 223. The packing effect of the baffle thread is proportional to the length of the threaded portion, speed of shaft rotation, and viscosity of the fluid being packed; it varies inversely with the height of thread grooves and greatly depends on the amount of clearance between the crests of the thread and the hole surface. The packing functions satisfactorily

if the radial clearance does not exceed 0.05-0.06 mm. Clearances over 0.1 mm render the packing inoperative.

Fine triangular thread (with a height of threads of 0.5 to 0.7 mm) offers the best sealing effect provided that the outside diameter is

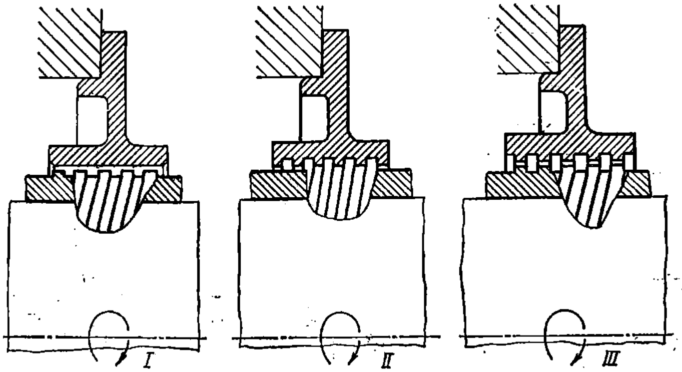


Fig. 221. Baffle-thread packings

ground to the size that ensures a small clearance in the assembly.

Square thread is only about half as effective. Trapezoidal thread takes an intermediate position. The optimum lead angle for triangular thread is from 5 to 10°, and for square thread, from 3 to 5° of arc.

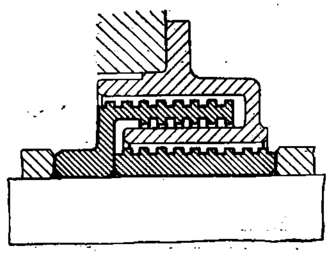


Fig. 222. Multi-level baffle-thread packing

The effectiveness of the baffle-thread packing may be improved through closing the thread grooves at their outlet with a cylindrical shoulder (Fig. 224).

In some designs, cylindrical springs with coils of round or square cross section are used in place of screw thread; the springs are mounted directly on the shaft or on an intermediate plain sleeve (Fig. 225, *I*, *II*) or else, in helical grooves cut in an intermediate casing (Fig. 225, *III*, *IV*).

The packing action of the spring is that its coils, brought up against the hole surface in the stationary flange sleeve, scrape from this surface the liquid that leaks from the housing. One end of the spring is fixed, and the other is free. It is essential to provide against self-jamming of the spring; with the direction of rotation such as shown in Fig. 225, the right-hand end of the spring must be fixed.

The spring packing is non-reversible and cannot be used in equipment where an accidental reversal of rotation is possible (e.g. in

speed reducers driven by asynchronous three-phase electric motors, where an erroneous phase may cause reversed rotation).

Drip-Fin Seals. These are used to break the oil film moving along the rotating shaft and throw the oil by the action of centrifugal

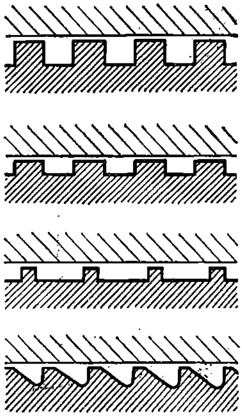


Fig. 223. Baffle-thread profiles

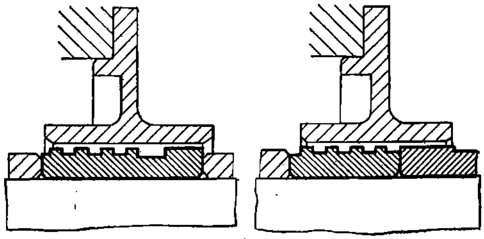


Fig. 224. Baffle-thread packing with a cylindrical shoulder

force into an annular cavity wherefrom it drains into the housing through drain passages.

The drip fins are made directly on the shaft (Fig. 226) or on appropriate removable parts (Figs. 227-229). Where rotational speeds

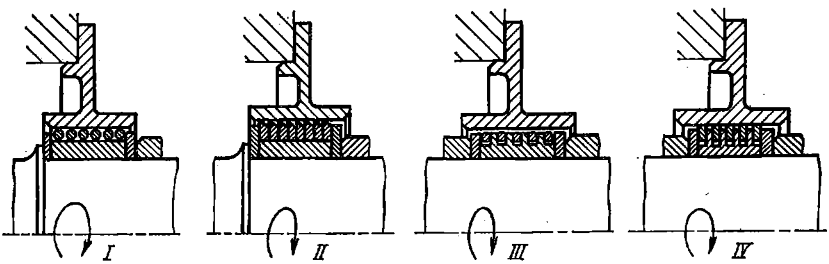


Fig. 225. Cylindrical spring packings

are not too high, a drip fin is replaced by a snap ring (see Fig. 227).

Baffle-Plate Packings. Baffle plates are located in front of gap seals in order to hinder the oil being packed from getting into the gap and to throw away the oil drops by the action of centrifugal force.

Fig. 230, *I-III* shows inefficient baffle-plate packings, which cannot prevent oil from working its way into the gap.

In the most effective design (Fig. 231, *I*), the housing sleeve is provided with a circular projection facing the baffle plate shaped as a cup whose brim overlaps the projection by a distance *a*. With this arrangement, oil can only get into the gap between the projection

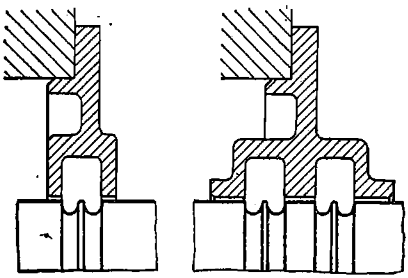


Fig. 226. Drip-fin seals

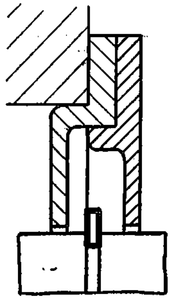


Fig. 227. Snap ring used as a drip fin

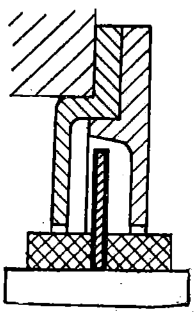


Fig. 228. Disc-type packing

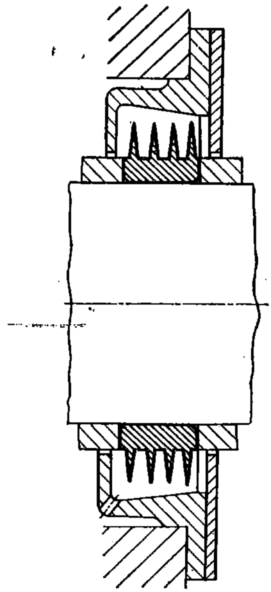


Fig. 229. Multi-fin packing

and the inner face of the housing; the dripping oil is taken up by the baffle plate and is driven away from the packing by centrifugal force. The reliability and simple design of this packing unit have promoted its wide application in the engineering industry.

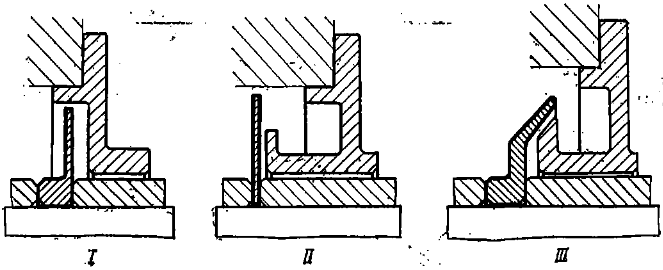


Fig. 230. Baffle-plate packings (inefficient designs)

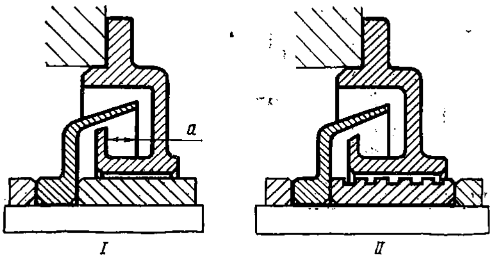


Fig. 231. Baffle-plate packings

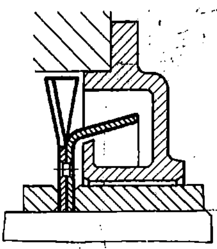


Fig. 232. Baffle plate coupled with axial impeller

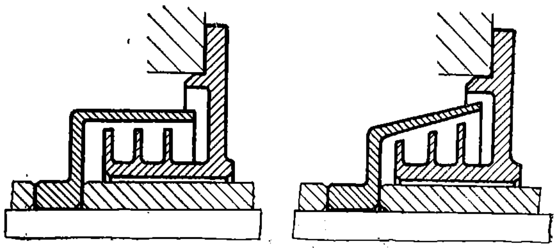


Fig. 233. Baffle-plate packing in combination with multiprojection sleeve

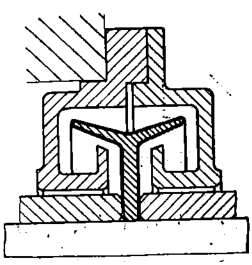


Fig. 234. Double-sided baffle-plate packing

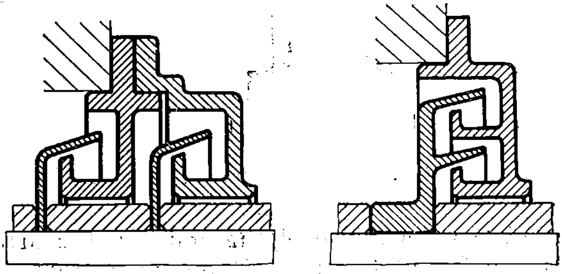


Fig. 235. Double baffle-plate packing arrangements

The circular gap between the shaft and the seal housing is often sealed additionally with labyrinth grooves or baffle thread (Fig. 231, *II*).

Figs. 232-237 illustrate constructions of baffle-plate seals. In the seal according to Fig. 232, the oil-baffling effect is enhanced by the use of a baffle plate coupled with a disc having helically-directed

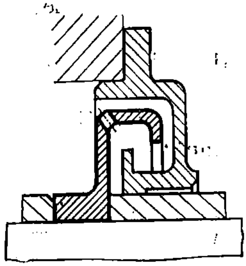


Fig. 236. Baffle plate with cylindrical rim

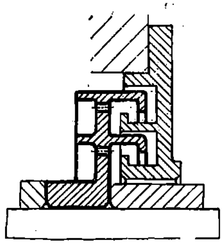


Fig. 237. Two-level baffle plate with cylindrical rims

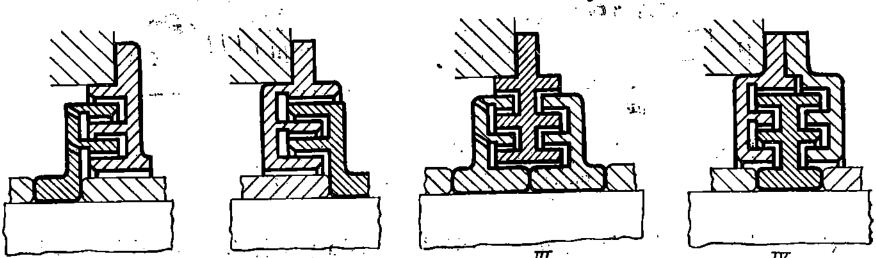


Fig. 238. Multi-level interlocked baffle plates

blades and acting as an axial impeller. The baffle plate in the seal in Fig. 236 is provided with a cylindrical rim. The oil coming into the circular groove formed by the rim is disposed of by the action of centrifugal force through a number of holes arranged around the plate periphery. Fig. 237 shows a double arrangement of this type.

Similar to the described constructions in the operating principle are the seals presented in Fig. 238, *I-IV*, which are conventionally referred to as axial labyrinth packings. They are, actually, multi-level baffle plates whose projections are interlocked with circular grooves in the stationary flanged sleeve, and their purpose is to remove the oil getting into the seal.

2.3. Axial Seals

Axial seals are of the sliding-contact type. The principle of an axial seal is shown in Fig. 239, *I*. Disc *a* is mounted on the shaft by a slide fit and held against rotation on the shaft by means of face

claws b. The disc is constantly spring-pressed against stationary washer c fastened to the housing.

The fluid being packed can leak through the seal in two directions indicated in the figure with arrows: through the disc face and through the annular clearance between the disc and the shaft. The leakage through the disc face is hampered by a tight contact between members a and c, and the leakage through the radial clearance, by rubber-ring packings d.

It is, therefore, apparent that an axial seal needs two packing elements: an *axial* and a *radial* one.

The radial packing operates under far better conditions than the axial packing because the disc displacements along the shaft are

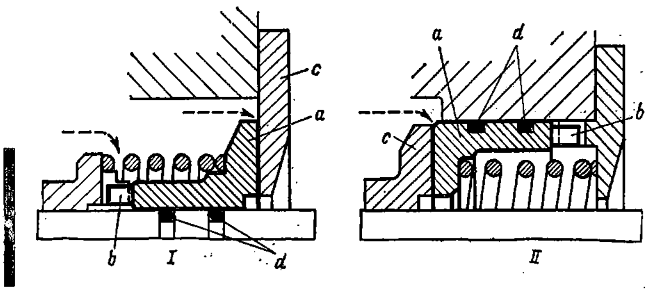


Fig. 239. Axial-seal principle. The axial packing disc is placed on shaft (I) and in housing (II)

extremely small. Here, any radial packing will be suitable—rubber rings, split metal rings, felt rings, lip packings, etc. The leakage through the radial clearance can be completely excluded by the use of a diaphragm packing, a bellows packing, etc. (see Figs. 243, 244). In an inverted-type axial seal (see Fig. 239, II), disc a is fixed against rotation relative to the stationary washer with face claws b. The disc is permanently spring-pressed against disc c installed on the shaft. Axial sealing is achieved by the contact between discs a and c, and radial sealing is effected by rings d.

The main advantage of axial seals is that the wear of the frictional surfaces is made up for by axial displacement of the spring-loaded packing disc. An axial seal wears in automatically; with the properly selected materials and slight lubrication, the frictional surfaces will be maintained in good condition for a long period of time, and the seal will function adequately.

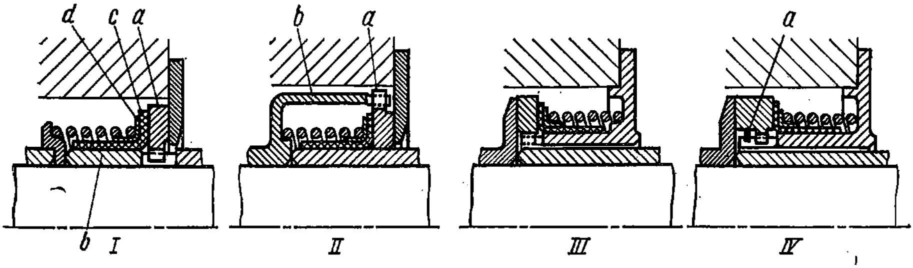
If the fluid being sealed is under pressure, the contact faces are loaded by this pressure as well as by spring pressure. Balanced axial sealing units are used for special applications.

Frictional surfaces are made of materials' combinations that provide antifriction properties: steel on babbit, hardened or nitrided steel

on bronze, graphite and coal compositions, plastics, etc. In critical applications, cast hard metal and metal ceramics are employed in combination with each other or with some of the mentioned softer materials. The frictional surfaces are machined to a roughness of 0.16 to 0.32 μm Ra.

For better sealing, frictional surfaces are provided with fine circular grooves (see Fig. 247).

The axially movable packing discs must be properly aligned on cylindrical guide surfaces to ensure strict squareness of the disc face



ig. 240. Axial seals with flange lip-type packings for tightening radial clearance

to the cylindrical surface and mutual parallelism of the movable and stationary discs. Self-aligning movable discs, which are employed in practice, secure more reliable contact on the sealing interface.

Fig. 240 shows some commonly used types of axial seals, wherein the radial clearance is tightened with rubber lip packings. In the seal according to Fig. 240, *I*, movable disc *a* is fixed against rotation on the shaft with claws made on the end face of intermediate sleeve *b*. A rubber L-packing *c*, serving as a radial seal, is tightly fitted over sleeve *b*; the packing face is spring-loaded through metal washer *d* to bear against the face of disc *a*. Axial displacements of disc *a* are ensured by the elastic properties of the packing. The disc is self-aligning.

Fig. 240, *II* presents a similar axial seal with the sleeve *b* of a different shape.

In the seal of an inverted type shown in Fig. 240, *III*, the packing disc is fixed relative to the housing. The snap ring *a* in the design presented in Fig. 240, *IV* makes the seal a self-contained unit that can be built as a whole into the housing. Fig. 241 illustrates a compact self-contained seal unit that makes use of a telescopic compression spring.

Fig. 242 presents axial seals with the radial clearance totally packed by means of a rubber bellows-type packing. This type of seal is suitable where there is no pressure in the chamber to be packed.

Axial seals wherein the radial clearance is packed with a metallic bellows are shown in Fig. 243; the movable discs are furnished with synthetic packing inserts. These designs are made self-contained with the use of a snap ring *a* (Fig. 243, *I*) or lock pin *a* (Fig. 243, *II*) introduced into a suitable slot in the disc. The bellows' ends are fixed, as usual, with the aid of rings *b*.

Fig. 244 shows a self-contained diaphragm-type axial seal, and Figs. 245 and 246, axial seals with felt-ring-type packings for sealing the radial clearance.

In the design according to Fig. 245, the packing rings are not compressed. Axial seals incorporating a spring-loaded packing unit

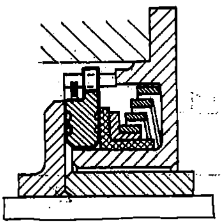


Fig. 241. Axial seal tightened with telescopic compression spring

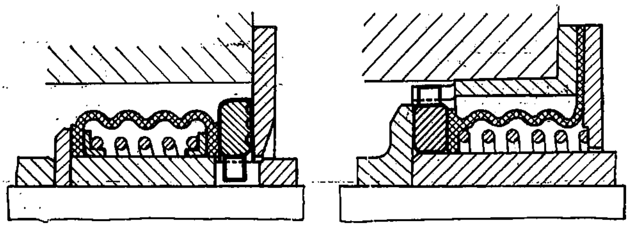


Fig. 242. Axial seal with a rubber bellows-type packing element for tightening radial clearance

of the stuffing-box type provide better performance (Fig. 246, *I*, *II*).

The radial clearance can also be packed in axial seals using lip packings (Fig. 247, *I*, *II*), rubber O-rings (Fig. 248, *I*, *II*), and split metal rings (Fig. 249).

Fig. 250 illustrates a balanced axial seal wherein the effect of fluid pressure is counteracted by spring pressure. The construction calls for the use of a stepped shaft; the larger and the smaller diameter of the step must be equal to the respective diameters in the movable packing member that limit the sealing area.

In some cases, satisfactory results can be obtained by the use of the simplest axial seals in the form of a plastic disc placed in a confined circular cavity and pressed against its walls by the action of the pressure differential on both sides of the seal (Fig. 251, *I*, *II*) or by spring pressure (Fig. 251, *III*). Fig. 252 shows multi-disc seals *I* and *II* of a similar type with the discs pressed together by spring pressure.

In an axial seal with a sleeve *a* fixed in the housing and spring-loaded (Fig. 253), radial sealing is effected by circular grooves made on the outside and inside surfaces of the sleeve.

Fig. 254 shows an axial seal using two packing sleeves pushed apart by spring pressure. While being free to move axially on the

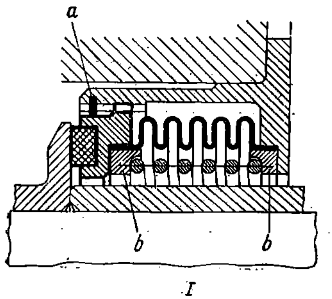


Fig. 243. Bellows-type axial seals

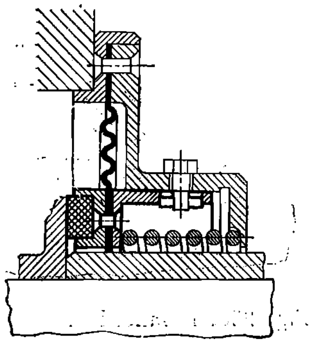


Fig. 244. Diaphragm-type axial seal

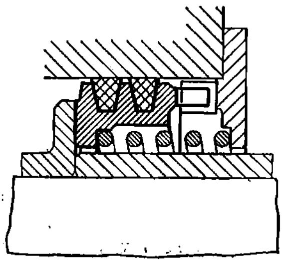


Fig. 245. Axial seal with groove packings for tightening radial clearance

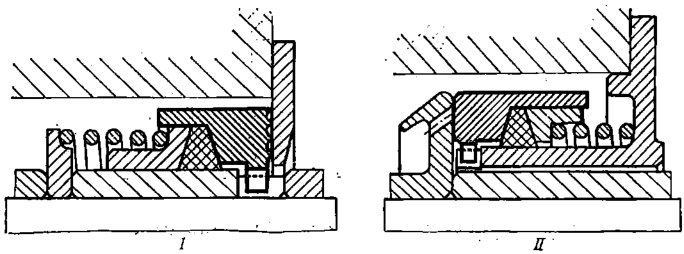


Fig. 246. Axial seals with stuffing boxes

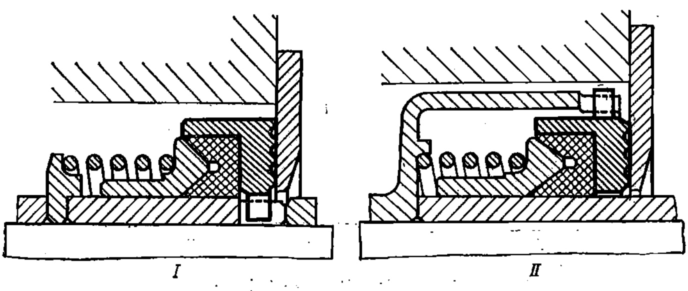


Fig. 247. Axial seals with lip-type radial packing elements

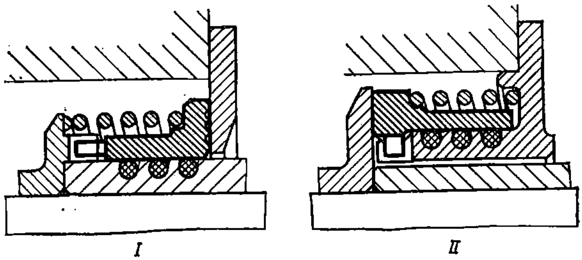


Fig. 248. Axial seal with rubber-ring packings

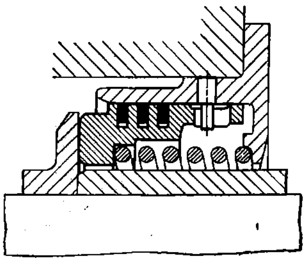


Fig. 249. Axial seal with split metal rings

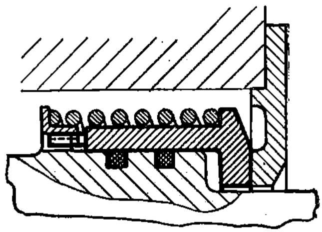


Fig. 250. Balanced axial seal

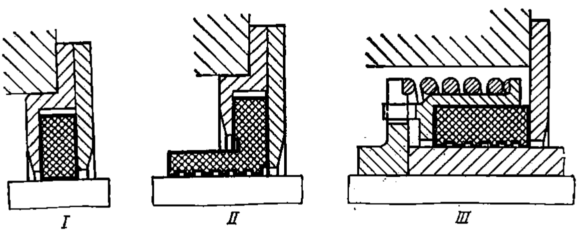


Fig. 251. Axial seals with a floating disc

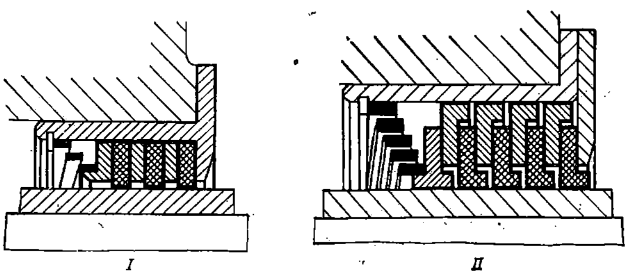


Fig. 252. Multi-disc axial seals

shaft, the sleeves are held against rotation relative to each other with suitable pins. A better design, presented in Fig. 255, has the

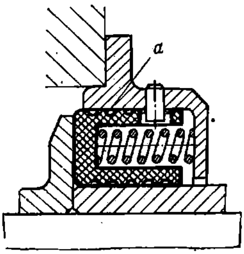


Fig. 253. Axial seal with a packing sleeve held in housing

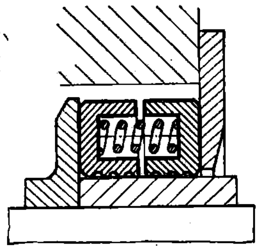


Fig. 254. Axial seal with floating spring-loaded sleeves

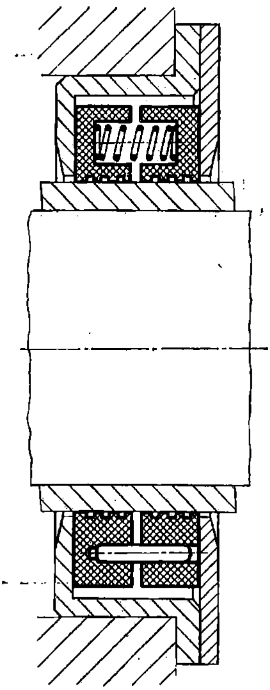


Fig. 255. Packing sleeves mounted in stationary housing

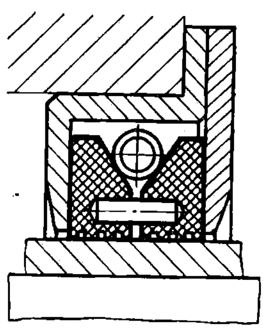


Fig. 256. Packing sleeves spring-loaded by garter spring

sleeves installed in a stationary circular case; the shaft is packed by the sleeves' cylindrical surfaces. A pressure lower than the fluid pressure is delivered to the right-hand sleeve.

In the seal shown in Fig. 256, the packing sleeves are forced apart by a garter spring that acts upon their outer tapered portions.

2.4. Combination Seals

The reliability of sealing may be improved if two or more seals of different types are installed in tandem. Some types of seals are

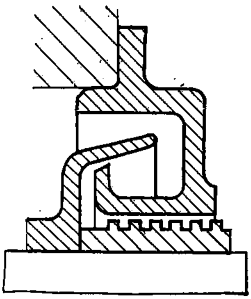


Fig. 257. Combination seal: baffle plate and thread

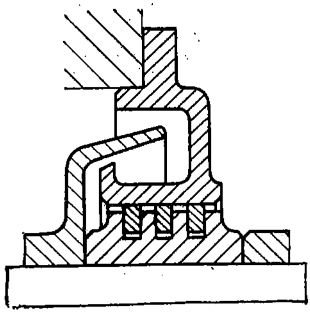


Fig. 258. Combination seal: baffle plate and split metal rings

well compatible and can be built into one and the same unit without any substantial increase in its overall dimensions.

It is common practice to combine a baffle plate with baffle thread (Fig. 257) or with split metal rings (Fig. 258). Also advisable is

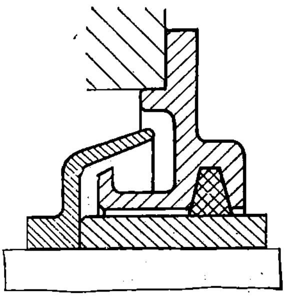


Fig. 259. Combination seal: baffle plate and felt-ring packing

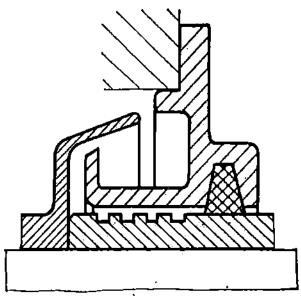


Fig. 260. Combination seal: baffle plate, baffle thread, and felt-ring packing

a combination of an oil trapping arrangement on the inside of a gap seal with felt packing rings (Figs. 259 and 260) or with a lip packing (oil seal unit) (Fig. 261) on its outside. The first sealing element traps the oil, and the second, prevents the penetration of dust and other foreign matter from the outside. Even a plain protective washer (Fig. 262) improves a seal and extends its service life.

Fig. 263 shows a combination of a baffle plate with an oil seal unit. The main feature of the design is that the packing is mounted on the baffle plate and rotates together with it; the sealing lip opera-

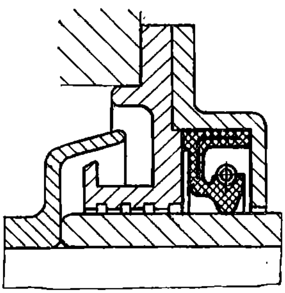


Fig. 261. Combination seal: baffle disc, circular grooves, and oil seal unit

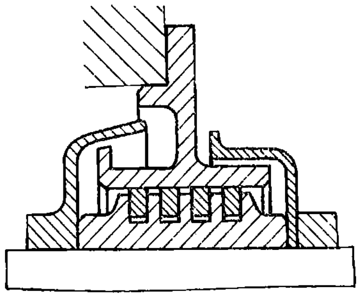


Fig. 262. Combination seal: baffle plate, split metal rings, and protective washer

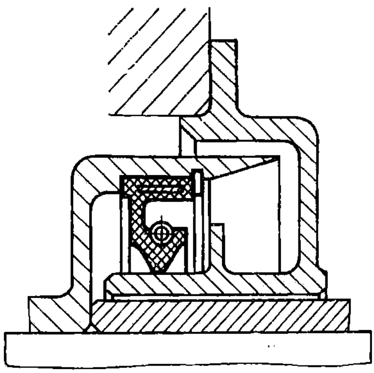


Fig. 263. Combination seal: baffle plate and oil seal unit

tes on the sleeve of the stationary oil-trapping housing. In this manner, adequate sealing is secured during both operation and shutdown of the machine. When the machine is at a standstill, sealing is effected by the lip packing, and when it is running, by the baffle plate: in running, centrifugal forces cause the packing to get loose on the sleeve and thus to become practically inoperative.

2.5. Air Seals

Oil-containing chambers are difficult to seal if the pressure in the chamber substantially exceeds the outside pressure (e.g. in the drive case adjoining the suction chamber in a centrifugal compressor).

Here, seals, even sliding-contact seals, cannot stop the leakage of oil from the high-pressure chamber into the low-pressure chamber. The oil mist penetrates into the low-pressure chamber through the seal.

A radical means for preventing the leakage is provided by two seals arranged in tandem and separated with an intermediate chamber into which compressed air is injected (*air seals*).

Fig. 264 exemplifies some versions of this type of seal.

Fig. 264, *I* shows diagrammatically a primary double-seal arrangement. The pressure in the chamber A exceeds that in the adjoining chamber B. The pressure differential causes the air carrying oil to leak from chamber A into chamber B. The diagram of pressures is shown at the bottom of the drawing.

In an air seal (Fig. 264, *II*), air at a pressure p_A equal to the pressure in chamber A is injected into the intermediate chamber between two seals. If chamber A is under the atmospheric pressure while chamber B is under vacuum, it will be sufficient to bring the intermediate chamber into communication with the atmosphere. It is apparent that the leakage of air through the left-hand seal will cease. Where there is no difference in pressures, any seals, both sliding and contactless, will prevent the ingress of oil into the intermediate chamber. Actually, leakage takes place through the right-hand seal, but here pure air without oil gets into chamber B out of the intermediate chamber.

An air seal is still more effective if compressed air is injected into the intermediate chamber at a pressure p_B exceeding the pressure in chamber A (Fig. 264, *III*). Here, the air leaks from the intermediate chamber into chamber A in the direction opposite to that of the possible leakage of oil. The air passes through the right-hand seal into chamber B. The drawback to this method is the need for a separate source of compressed air.

Figs. 265 and 266 present instances of air-seal designs.

A logical consequence of the atmospheric-air sealing is complete separation of the chambers sealed with an air space. This method is widely used for bearing assemblies situated near vacuum chambers. Bearing units in separate housings are placed at some suitable distance from a vacuum chamber; the bearing housings and the vacuum chamber are sealed independently with conventional sliding-contact or contactless packings.

2. 6. Labyrinth Seals

Labyrinth seals are used to tighten chambers filled with gas and steam. Their action consists in that the fluid being sealed slows down its escape in a narrow annular gap due to the formation of vortices

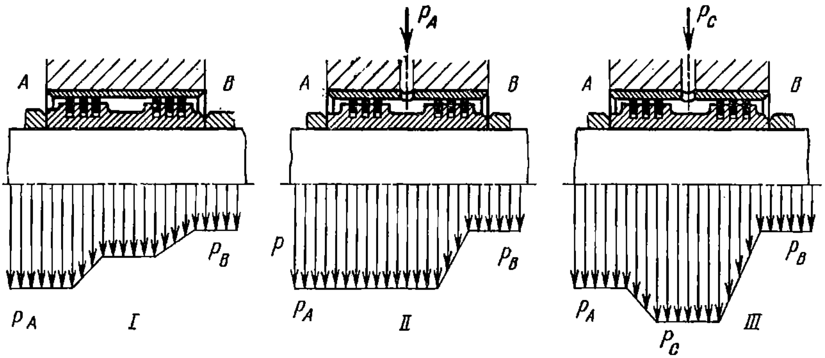


Fig. 264. Principle of air seal

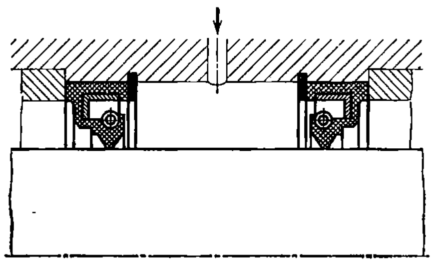


Fig. 265. Air seal using lip packings

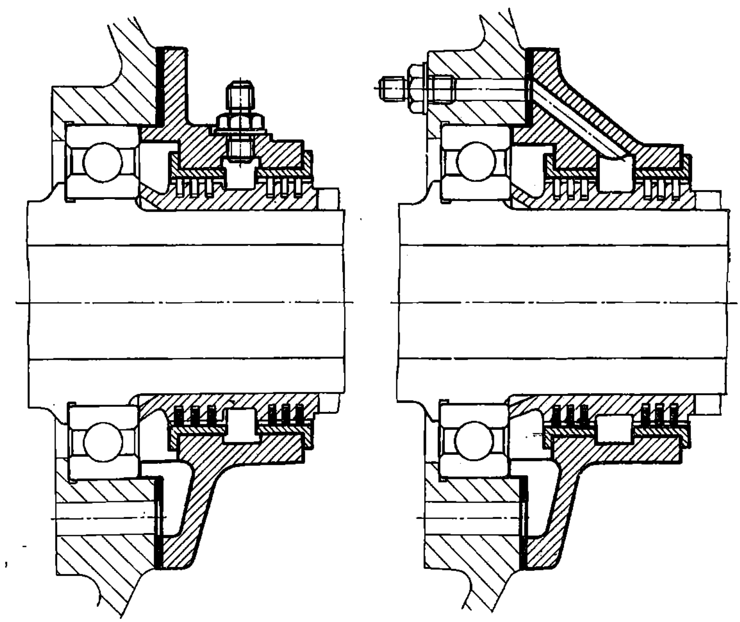


Fig. 266. Air seal for antifriction bearings

and subsequently expands in an adjoining annular pocket which has a much larger volume. In the annular gap, the fluid pressure is transformed into a velocity head; the pressure of the fluid escaping from the gap is restored only in part, while its other part is expended in irrecoverable losses occurring in the process of the vortice formation and expansion. The higher these losses (or, in dimensional terms, the sharper the edges forming the gap and the smaller its cross section), the lower the portion of the pressure that is recovered in the pocket and, hence, the more effectively the seal operates.

By arranging a number of pockets separated with narrow gaps in a series, leakage through the seal can be substantially decreased.

Labyrinth seals are used for tightening shafts operating at high peripheral speeds and fluid pressures, where sliding-contact packings are unsuitable. Labyrinth seals can function practically at unlimited rotational speeds and pressures.

The principle of a labyrinth seal is explained in Fig. 267. The seal separates chamber A from chamber B, which are, respectively, under the higher pressure p_A and the lower pressure p_B.

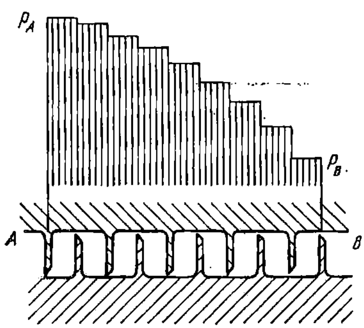


Fig. 267. Principle of labyrinth seals

The fluid escapes through the first annular gap at a high speed, which drops almost to zero in the contiguous annular pocket. Because of pressure losses that stem from the formation of vortices in the gap, a pressure lower than that in chamber A is created in the pocket. Since the gas specific volume is higher in the pocket than in chamber A, while the volume of gas escaping in a unit time is the same in virtue of the continuity of flow, the speed of the fluid in the second annular gap is bound to be higher than in the first one, and, similarly, higher in each succeeding gap than in the preceding one. As a result, the difference in pressure between two adjoining pockets rises stepwise over the whole length of the seal.

At higher pressure differentials and with a large number of steps, a critical difference of pressures can arise in one of the gap portions; the speed of the fluid will reach there the sonic level. In such a seal, all subsequent steps are unnecessary as they cannot reduce the critical flow equal to the product of the sound velocity by the gap cross-sectional area. The number of steps in a labyrinth seal is found by the appropriate thermodynamic calculation.

A labyrinth seal cannot completely exclude the escape of gas. Moreover, continuous gas flow through the labyrinth forms the base

of its operating principle and is the necessary condition for the action of this seal. The labyrinth can only reduce the fluid escape.

The only exception is the case where pressure in the chamber being packed varies cyclically from maximum to zero. Here, the wave of gas entering the labyrinth possesses a limited quantity of energy, which can be completely expended in the seal. Under such

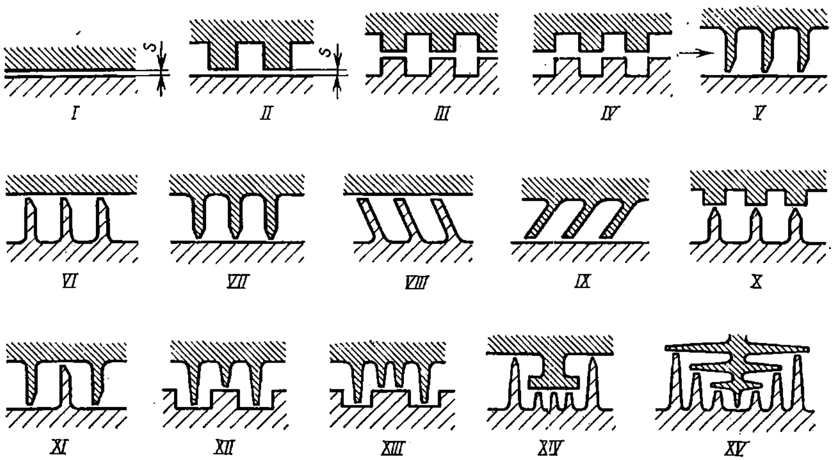


Fig. 268. Types of labyrinths (here and in Figs. 269-271 labyrinth cross sections are shown)

conditions, labyrinth seals are capable of ensuring complete tightness.

Fig. 268 illustrates some varieties of labyrinth seals (in order of increasing effectiveness). Fig. 268, *I* presents a plain sealing gap; the introduction of projections (Fig. 268, *II-IV*) substantially reduces gas escape (by 50 to 65%) over the same sealing length and the same minimal gap *s*.

In the labyrinths according to Fig. 268, *II-IV*, the axial extension of the seal is utilized inadequately. Thin and high disc-type partitions that allow a greater number of pockets of a desired volume to be arranged over a unit length are more preferable. In addition, thin sharp-edged walls, causing greater vortex losses, make for more effective sealing.

Fig. 268, *V* shows partitions made in the housing, and Fig. 268, *VI* those made on the shaft. The partitions are beveled sharp against the direction of flow. Fig. 268, *VII* shows partitions with a double-sided chamfer that are adapted for sealing from both sides. A further improvement in efficiency is achieved with the use of partitions slanting towards the direction of the gas flow (Fig. 268, *VIII*, *IX*).

The design according to Fig. 268, *IX* with the slanting partitions made in the housing has an important advantage: the partitions coming into an accidental contact with the shaft get warm and, consequently, come off the shaft, so preventing further worsening of the defect.

Fig. 268, *X* illustrates a design wherein thin and wide projections are combined. This type of labyrinth seal can be used for shafts fitted into the housing both radially and axially. The radial assembly to housings having a diametral parting plane offers greater possibilities for the development and use of labyrinth seals. Fig. 268, *XI*

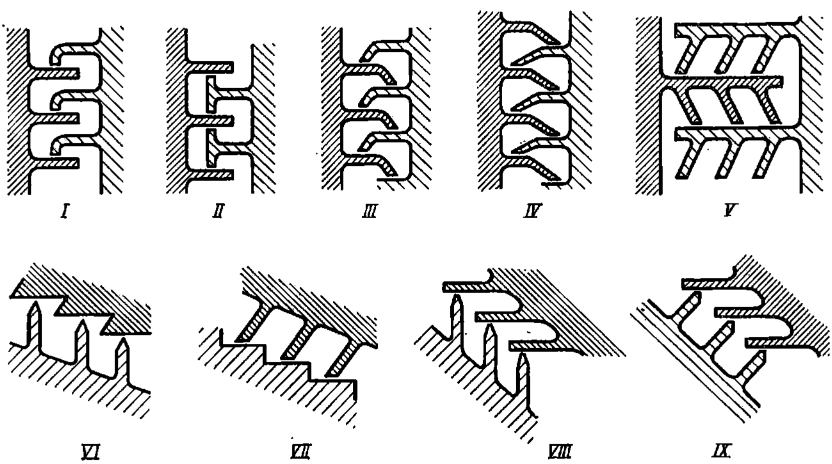


Fig. 269. Disc-type labyrinths

shows a labyrinth wherein the partitions on the shaft and in the housing are interlocked; here, the gas flow repeatedly changes its direction, whereby the sealing effect is increased. Some varieties of complex-pattern labyrinths are illustrated in Fig. 268, *XII—XV*; these designs require radial assembly.

In an axially restricted space, the labyrinths are developed in a radial direction. They are made up of two discs, one rotary and the other stationary, which are provided with interlocking face partitions (Fig. 269, *I*, *II*). In the designs according to Fig. 269, *III*, *IV*, the partitions have the property to recede, when warmed, from their counterparts. The seal in Fig. 269, *V* extends in both radial and axial directions. Slanting labyrinths (Fig. 269, *VI-IX*) consist of two conical discs with stepped ridges or thin partitions. In the constructions according to Fig. 269, *VII-IX*, the partitions are of the receding type.

The most effective sealing is achieved where the gap between the partitions and the shaft is kept to a minimum; this gap, however,

cannot be smaller than the total amount including the radial clearance in the shaft bearings, the amount of their out-of-alignment with the seal housing, and the amount of deflection of the shaft in operation. The radial gap in small- and medium-diameter seals is practically held to 0.05 to 0.2 mm.

The hazard of damage of a labyrinth seal due to a contact of its stationary and rotary members during assembly is prevented by

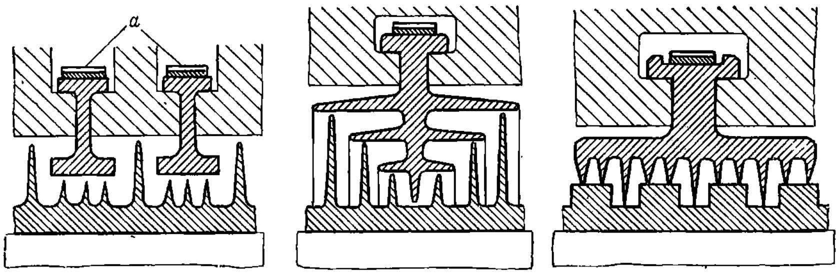


Fig. 270. Installation of labyrinths in split housings

a method illustrated in Fig. 270. The stationary part of the labyrinth seal consists of several sectors each provided with a T-shank introduced into an appropriate annular T-slot in the housing; the sectors are pressed against the slot's cylindrical surface by leaf springs *a*. When catching on the shaft, the sectors overcome the spring pressure and retract somewhat, thus keeping the partitions safe from damage.

In some seal constructions, the edges of the partitions are made very sharp (about 0.1-0.2 mm wide), and the radial gap is made intentionally smaller than required in order that it may take the needed amount by itself as a result of crushing and burning of the edges from contact with the shaft. If the partitions are sufficiently thin and made from a soft metal, whereas the shaft is surface-hardened, the latter will get no damage in the process, and the minimum gap needed to meet actual operating conditions will develop automatically.

Fig. 271 presents some methods of securing the partitions in their housings. In the designs according to Fig. 271, *I*, *II*, the partitions with spacer sleeves and the Γ-shape partitions, respectively, are secured in the housing by upsetting the ends of the latter (these seals are assembled axially). The half-circular partitions shown in Fig. 271, *III* are fitted into annular grooves of a split housing. Fig. 271, *IV* shows a method of securing the partitions in a housing, produced from a ductile metal, by upsetting the housing's material; in the designs according to Fig. 271, *V*, *VI*, the partitions are wedged in the housing with upset annular or segmental inserts of soft metal;

and in the designs illustrated in Fig. 271, *VII*, *VIII*, stamped partitions are fixed in the housing by upsetting soft-metal wires. The seals presented in Fig. 271, *III-VIII* are assembled radially.

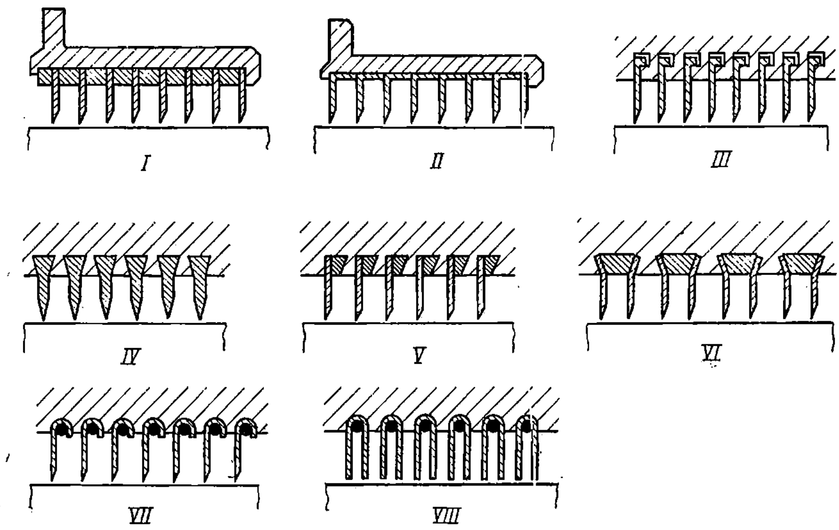


Fig. 271. Methods for securing partitions in housings

2.7. Centrifugal Fluid Seals

A centrifugal fluid seal (Fig. 272) consists of an impeller rotated in an enclosed annular chamber wherein a sealing liquid (oil, water, etc.) is contained. Centrifugal force urges the liquid against the chamber periphery. If some pressure acts upon the sealing liquid from one side, the liquid will take the position shown in Fig. 272. The difference of centrifugal forces acting upon the liquid on both sides of the impeller determines the pressure the seal can withstand (in kgf/cm^2):

$$p = 10^{-7} \frac{\omega^2 \gamma}{4g} (D_2^2 - D_1^2)$$

where ω = angular speed of rotation for impeller, s^{-1}; γ = density of sealing liquid, kgf/dm^3; g = acceleration of gravity (g = 9.81 m/s^2); D_2 and D_1 = diameters of liquid rings on both sides of impeller, cm.

The maximum pressure resisted by the seal (the extreme case where almost all the liquid is shifted to one side of the impeller)

$$p_{\max} = 10^{-7} \frac{\omega^2 \gamma}{4g} (D_0^2 - d_0^2)$$

where D_0 and d_0 = outside and inside diameters of impeller, respectively.

A minimally required liquid volume to be contained in the seal

$$Q=\frac{\pi}{4}(D_0^2-d_0^2)\,b$$

where b = width of impeller blades, cm.
Hence,

$$p_{\max}=10^{-7}\frac{\omega^2\gamma}{\pi g b}Q$$

Centrifugal fluid seals can withstand considerable pressures. For example, a seal with an impeller about 200 mm in diameter, operating on oil as a sealing liquid, will stand up to an excess pressure about 3 kgf/cm^2 at a rotational speed of 2,000 rpm. With the use of heavy liquids, e.g. mercury, the allowable working pressure can be raised in the considered example up to 50 kgf/cm^2.

A centrifugal seal must be provided with a system for circulation of the sealing liquid and removal of heat arising from the impeller rotation. Otherwise, the sealing liquid will quickly overheat in operation.

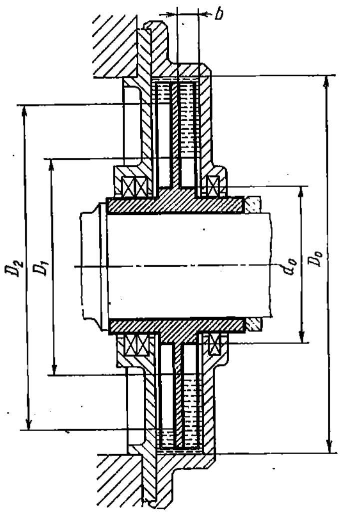


Fig. 272. Principle of centrifugal fluid seal

2.8. Sealing of Reciprocating Parts

2.8.1. Rod Packings

Reciprocating parts, such as piston rods or reciprocating-pump rams, are sealed mostly by means of stuffing boxes (see Figs. 273 and 194), with packing materials suited to operating conditions.

Piston rods in hydraulic, pneumatic and vacuum cylinders working at low pressures and temperatures are packed with O rubber rings fitted into suitable grooves cut in the end wall of a cylinder (Fig. 274).

For operation at high pressures and temperatures, use is made of stuffing boxes with expansion metal rings (Fig. 275). The seal consists of a set of alternating rings with inner and outer tapers. When tightened, the inner-taper rings expand and come to bear against the housing-hole surface, and the outer-taper rings contract, packing the rod surface.

The inner-taper rings must be more elastic than the outer-taper rings and come into contact with the surface of the housing hole before the clearance between the outer-taper rings and the piston rod is made up for. The clearance depends on the extent of tightening. If no proper care is taken, the seal may be easily over-tightened to jam the rod completely.

To prevent self-jamming of the rings, half the included angle of the taper (Fig. 276) must be larger than the angle of friction. In the constructions employed, α is from 12 to 20° of arc.

Figure 277, *I-III* shows different forms of expansion rings. The rings according to Fig. 277, *III* have improved elasticity. In some

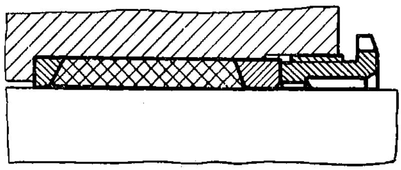


Fig. 273. Piston rod tightened with stuffing-box packing

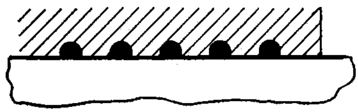


Fig. 274. Piston rod tightened with rubber rings

types of rod seals, better elasticity is achieved by making the rings split; the use of such rings, however, reduces the sealing effect.

The expansion rings are produced from spring steel; they are subjected to hardening and medium tempering. For use in critical applications, the rings are made from berillium bronze.

Segmental-Cut Ring Packings. Segmental-cut rings are metal packing rings divided circumferentially into several (usually, three)

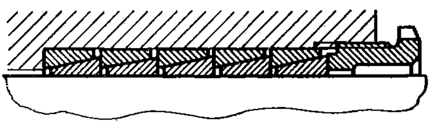


Fig. 275. Stuffing box with expansion metal rings

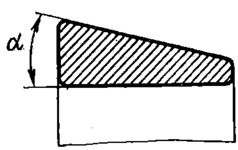


Fig. 276. Half the included angle of taper in expansion metal rings

sections. This type of packing is difficult to produce and requires careful assembly; however, it is reliable and capable of withstanding high pressures and temperatures.

The ring (Fig. 278) consists of three sections connected through step joints and held together with a garter spring. The assembled ring is encased in a circular shell and mounted on the rod. Under the action of fluid pressure in the chamber being packed, the ring is pressed on one side against the housing wall; radial sealing is

effected through tightening the ring assembly by the garter spring. For better sealing, labyrinth grooves are cut in the ring's inner

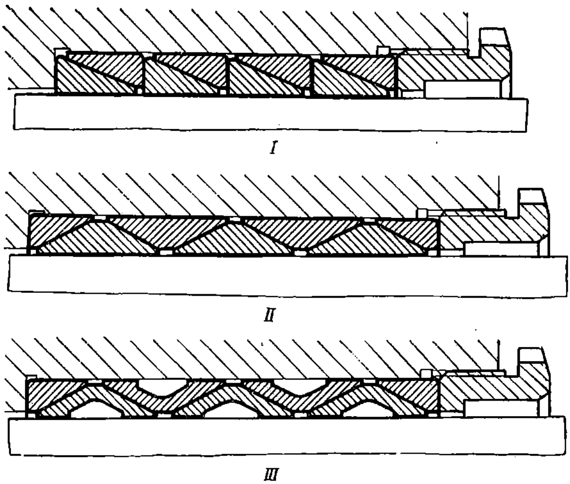


Fig. 277. Forms of expansion rings

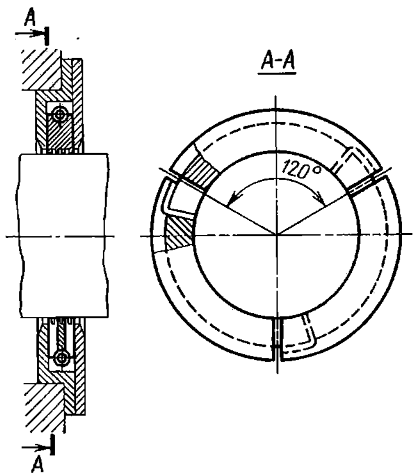


Fig. 278. Segmental-cut packing ring held together by garter spring

surface. A number of segmental rings, arranged in tandem, are normally used.

Figure 279 shows a set of two segmental rings mounted in a com mon shell. The rings are secured to each other with pins in such

a way that the joints between the segments on one ring face integral portions on the other ring; in addition, the rings are fixed against

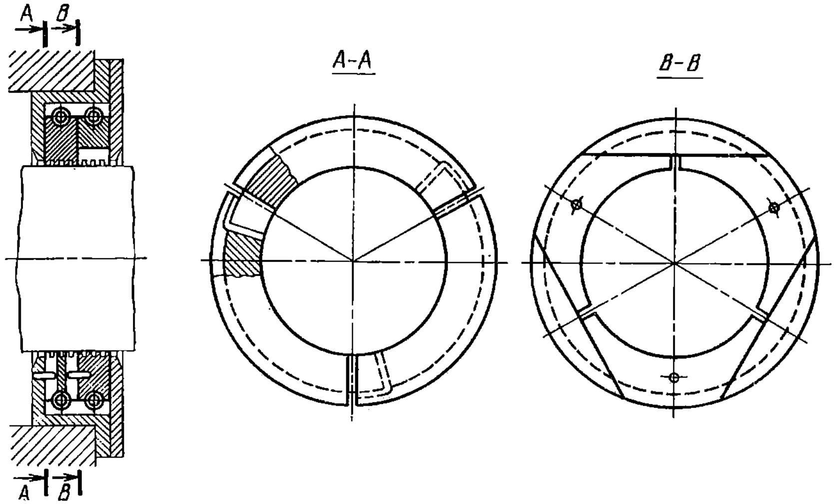


Fig. 279. Double arrangement of segmental rings in a common shell

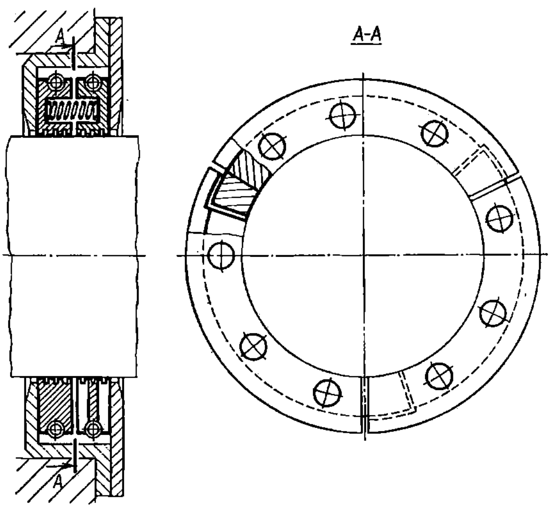


Fig. 280. Segmental rings freely turnable in their shell

rotation in the shell (such packings are also applicable to sealing rotary shafts).

In the design according to Fig. 280, the rings are held together with the aid of pins and are free to rotate in the shell; axial sealing

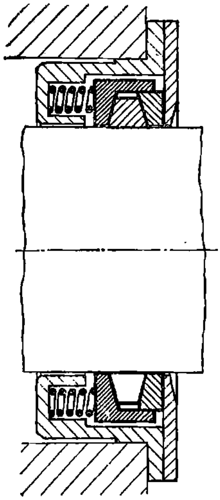


Fig. 281. Spring-loaded conical segmental rings

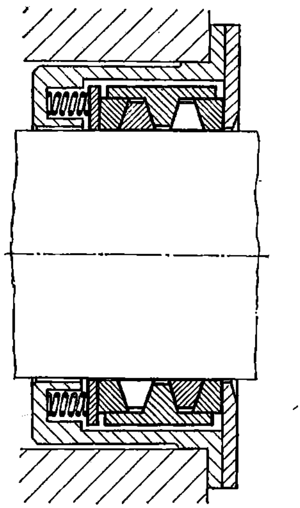


Fig. 282. Arrangement of two conical segmental rings in tandem

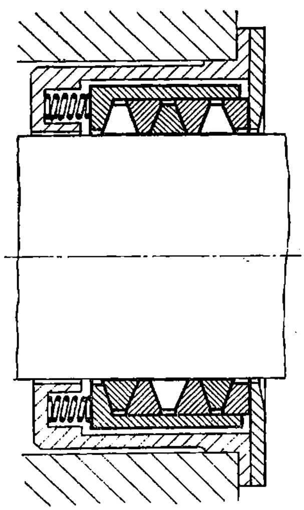


Fig. 283. Arrangement of three conical segmental rings in tandem

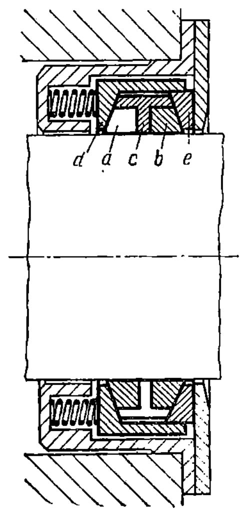


Fig. 284. Segmental-ring packing unit

is effected through pressing the rings against the respective shell end walls by compression springs. Double-ring arrangements similar to those shown in Figs. 279 and 280 can withstand an excess pressure of 10 to 15 kgf/cm^2.

The seal design in Fig. 281 features a segmental ring of a V-cross section; the spring is housed by two shells and pressed against the rod with compression springs.

Figs. 282 and 283 illustrate sealing arrangements with sets of two and three rings, respectively.

In the design shown in Fig. 284, segmental V-section rings *a* and *b* are located in split T-ring *c*; the ring set is compressed with tapered retainers *d* and *e*. The arrangement of several sets of this type in a series can withstand pressures of several hundreds of atmospheres.

The rings are produced from wrought bronze; for some applications, they are lined with babbitt. The appropriate rod and housing surfaces must be hardened (or nitrided) and finished to a roughness of 0.63-1.25 μm Ra. Frictional surfaces should be moderately lubricated.

2.8.2. *Piston Packings*

Small-diameter pistons (plungers in hydraulic pumps, oil fuel pumps, etc.) are sealed by lapping to their cylinders. The sealing effect is improved by introduction of labyrinth grooves (Fig. 285).

Large-diameter pistons operating at low temperatures and pressures (e.g. in hydraulic, pneumatic and vacuum cylinders) are sealed with labyrinth grooves or O rubber rings (Fig. 286, *I*, *II*). At high fluid pressures, lip packings are used (Fig. 286, *III*). The most dependable and universal piston packings, capable of functioning at high temperatures and withstanding the highest pressures are piston rings (Fig. 286, *IV*); they are used for operation against both liquids and gases.

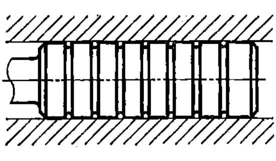


Fig. 285. Lapped plunger with circular labyrinth grooves

Piston Rings. A piston ring is a split metal ring (normally of a square cross section), located in an appropriate circular groove cut in the piston. The outside diameter of a piston ring in the free condition is larger than the cylinder-bore diameter. When introduced into the cylinder, the piston ring contracts, and, owing to its resilience, becomes tightly pressed against the cylinder-bore surface around its circumference, except for the slit portion forming a narrow passage.

In operation, piston rings are urged against the cylinder walls not only by their inherent resilience, but also by the pressure of the fluid entering the piston-ring grooves and acting upon the inside

peripheral surface of the ring (Fig. 287). This pressure may be many times that caused by the ring's resilience and is predominant in producing the sealing effect. The tightness of the piston ring inserted

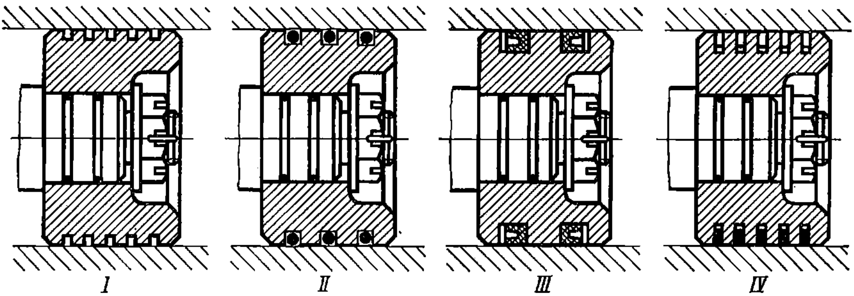


Fig. 286. Piston packing methods

into the cylinder is only a preliminary condition for this pressure to arise.

In this respect, piston rings are similar in action to lip packings: both packing elements are pressed against the cylinder bore surface by a force proportional to the fluid pressure.

On the other hand, piston rings act similarly to a labyrinth seal. The rings are placed in piston grooves with an axial and a radial

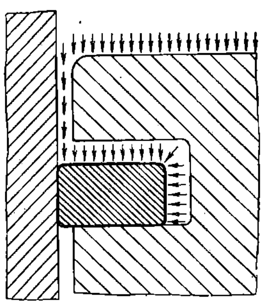


Fig. 287. Action of fluid pressure on piston ring

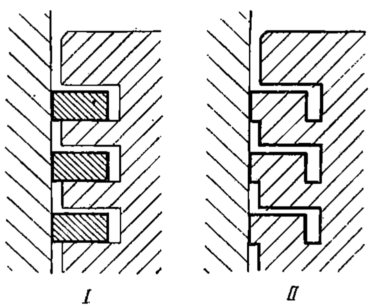


Fig. 288. Packing principle of piston rings

clearance (Fig. 288, *I*). Pressed against the walls of the piston grooves, the rings form a series of annular pockets (Fig. 288, *II*). The working fluid, entering the pocket at the first piston ring, can only pass into the next annular pocket through the narrow slit in the piston ring. As the fluid goes through the slit, its pressure drops; the process repeats at every subsequent piston ring. As a result, the fluid pressure in the last pocket will be substantially lower than in the first pocket.

As a rule, the fluid pressure in the cylinder chamber varies cyclically from a maximum (on the working stroke of the piston) to zero (on the return stroke); the fluid wave coming into the seal possesses a limited quantity of energy, which can be completely expended as the fluid is passing through the seal. Under such conditions, a labyrinth seal can provide complete tightness.

For more reliable sealing, several (usually three) piston rings are employed. In operating against high-pressure fluids, use is made of 5 to 10 and, in some applications, even more piston rings.

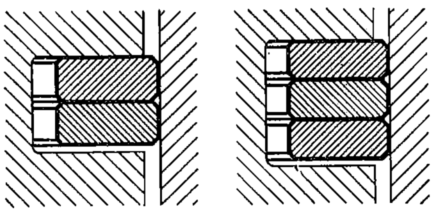


Fig. 289. Several piston rings placed in one groove

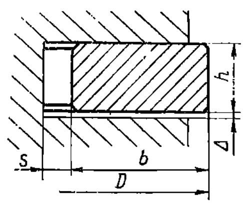


Fig. 290. Design-clearances between piston ring and its groove

To save space in the axial direction, two or more rings are placed in one piston groove (Fig. 289).

Piston rings are fitted into the grooves with an axial clearance Δ about 0.05 to 0.1 h (Fig. 290).

Clearance s between the inner periphery of a piston ring and the groove bottom should be held to within 0.2 to $0.25b$.

The joint slit is chosen in such a way that the piston ring brought within the cylinder still has a clearance in the slit that prevents the closing of the joint. This clearance should be held to a minimum in order to reduce the fluid escape through the joint and to provide an allowance for rapid increase of the clearance due to wear of the piston ring and the cylinder bore (the amount of clearance increases as $2\pi\delta$, where δ is radial wear of both piston ring outside periphery and cylinder-bore surface).

In practice, the clearance is specified at (0.002 to 0.005) D_0, where D_0 is the cylinder-bore diameter.

For higher-temperature applications (e.g. in compressor and engine cylinders), the clearance should be increased by Δ_t, which is the ring thermal elongation found from the relationship

$$\Delta_t = \pi D_0 [\alpha_r (t_r - t_0) - \alpha_c (t_c - t_0)]$$

where α_r and α_c = coefficients of linear expansion for piston ring and cylinder materials, respectively; t_r and t_c = working temperatures for ring and cylinder, respectively, °C; t_0 = initial (or assembling) temperature, °C.

Strength Calculation for Piston Rings. The piston ring diameter in the free condition is so selected as to ensure adequate tightness between the ring and the cylinder bore surface in assembly. At the same time no high stresses are permissible in the ring's material both in operation, when the ring is compressed by the cylinder walls, and in fitting the ring into its groove in the piston, when the ends of the ring at the joint are pulled apart for the assembly with the piston. The critical section lies on the axis of symmetry of the ring in the opposite relation to the joint. During operation, the outer fibres of the section are subjected to tension, and the inner fibres, to compression; during fitting the ring in place, the outer fibres are compressed, and the inner, expanded.

The maximum working stress

$$\sigma_{max} = E\frac{b}{D_0}\left(1 - \frac{D_0}{D}\right) \tag{3}$$

where D_0 = diameter of cylinder bore, mm; D = outside diameter of ring in free condition, mm; b = width of ring section, mm; E = modulus of elasticity of ring material, kgf/mm^2.

Hence,

$$\frac{D}{D_0} = \frac{1}{1 - \frac{\sigma_{max}}{E}\frac{D_0}{b}} \tag{4}$$

The maximum stress in the ring while being fitted over the piston

$$\sigma'_{max} = 2E\left(\frac{b}{D_0}\right)^2 - \sigma_{max} \tag{5}$$

As the stress σ'_{max} is only effective for a short time, i.e. when placing the ring, its magnitude may be allowed to exceed that of working stress σ_{max}, which acts constantly; the latter must be lower than the former also because the ring normally operates at an elevated temperature.

Let us assume that $\sigma'_{max} = a\sigma_{max}$, where a is larger than unity (on the average, a = 1.5-2.0).

Substituting this expression into equation (5), we shall have

$$\sigma_{max} = \frac{1}{1+a}2E\left(\frac{b}{D_0}\right)^2 \tag{5a}$$

whence,

$$\frac{\sigma_{max}}{E} = \frac{2}{1+a}\left(\frac{b}{D_0}\right)^2$$

Substituting the expression of σ_{max}/E into equation (4), we obtain

$$\frac{D}{D_0} = \frac{1}{1 - \frac{b}{D_0}\frac{2}{1+a}} \tag{6}$$

Equation (5*a*) will give

$$\frac{b}{D_0}=\sqrt{\frac{\sigma_{\max}(1+a)}{2E}} \tag{7}$$

Table 1 lists values of b/D_0 calculated by equation (7) for piston

Table 1

Values of b/D_0 for Piston Rings of Different Materials

Material	σ_{max}, kgf/mm²	E, kgf/mm²	b/D_0
Cast iron	12	8,000	1/21
Steel	35	22,000	1/20
Bronze	15	11,000	1/22

rings, made from different materials, at common values of $\sigma_{max}(a=2)$.

From the table it follows that the values of b/D_0 for all three materials do not differ significantly. On the average, b/D_0 may be assumed to be equal to 1/20. Substituting this value and the value of $a = 2$ into equation (6), we shall obtain $D/D_0 = 1.035$.

For any specific application, ratios b/D_0 and D/D_0 should be determined with equations (7) and (6), substituting appropriate values of σ_{max} and a.

The pressure of the piston ring on the cylinder walls (assuming that the pressure around the ring circumference is spread uniformly) is

$$p=\frac{\sigma_{max}}{3}\left(\frac{b}{D_0}\right)^2$$

Taking b/D_0 to be equal to 1/20 and substituting the above values of σ_{max}, we shall obtain the following average values of pressure for the rings of various materials: 1 kgf/cm² for cast iron, 3 kgf/cm² for steel, and 1.25 kgf/cm² for bronze.

The foregoing relationships make it possible to formulate the following design requirements for round piston rings: (1) the ring width b should not exceed 1/20 the cylinder-bore diameter; (2) the ring diameter in the free condition should not be larger than 1.03-1.04 the cylinder-bore diameter.

If taken in excess of the stated values, these parameters will cause high stresses in both operation and fitting the ring onto the piston. This excess should be well founded by the calculation for any specific case. If, for some reason, piston rings with a ratio $b/D_0 > 1/20$ (e.g. in small-cylinder applications) need to be used,

they should be mounted on removable intermediate washers (Fig. 291). Height h of the ring (Fig. 290) does not influence the magnitude of stress in it and the pressure it exerts on the cylinder-bore surface.

An increased height only results in higher stiffness, which will diminish the radial sealing effect and increase the force required to mount the ring on the piston.

Height h of the ring is normally taken to be $(0.5 \text{ to } 0.7)b$. To find the height, use may be made of a relationship

$$h = 2 + (0.01\text{-}0.03)\, D_0$$

where D_0 = cylinder-bore diameter.

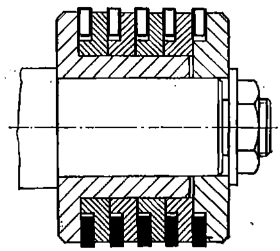


Fig. 291. Piston rings mounted on removable washers

Uniform-Pressure Piston Rings. Round piston rings provide no uniform pressure along their circumference. A typical pressure distribution diagram for the round rings is shown in Fig. 292.

Uniform pressure could be obtained if the rings were formed with two eccentric circles making contact with each other (Fig. 293, *I*). Such rings, however, are not feasible; they can only be approximated to some extent (Fig. 293, *II*). This

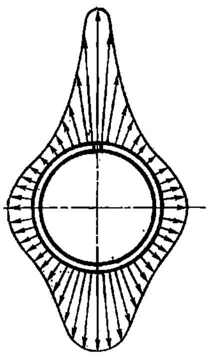


Fig. 292. Diagram of pressure distribution along the circumference of round piston rings

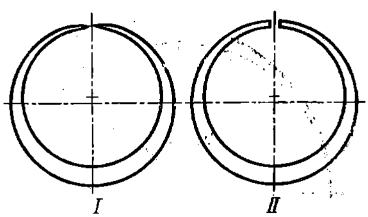


Fig. 293. Piston-ring contour for uniform pressure along circumference

latter form is sometimes given to snap rings to distribute the pressure more uniformly around the circumference and improve the ring's resilience for easier fitting in place.

Another method of obtaining the uniform pressure consists in that the ring is made so as to have a form slightly resembling an ellipsis in the free condition (these rings are conventionally called

"elliptical"). The ring introduced into the cylinder takes a round form and exerts a uniform pressure on the bore surface.

The coordinates of the axial line for a uniform-pressure piston ring (Fig. 294) are found from the relationships:

$$X=\frac{\sigma_{max}}{4E}\,\frac{D_0^2}{b}\,A_x \qquad (8)$$

$$Y=\frac{\sigma_{max}}{4E}\,\frac{D_0^2}{b}\,A \qquad (9)$$

where A_x and A_y are dimensionless quantities depending on angle ψ only,

$$A_x=1-\cos\psi-\psi\sin\psi-\frac{\sin^2\psi}{2}$$

$$A_y=\frac{\psi}{2}+\sin\psi-\psi\cos\psi-\frac{\sin\psi\cos\psi}{2}$$

The values of A_x and A_y are represented in Fig. 295 as a function of angle ψ. Stress σ_{max} in equations (8) and (9) is determined by strength calculations with equation (3).

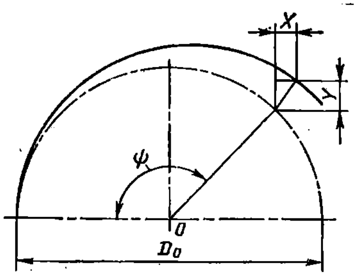


Fig. 294. Determining axial line coordinates for uniform-pressure piston rings

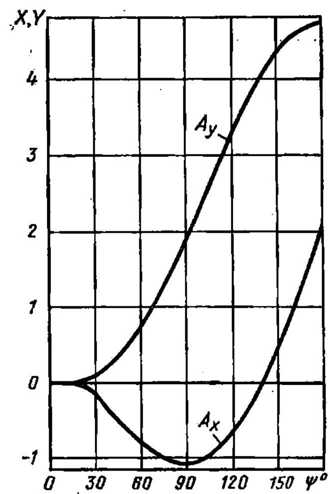


Fig. 295. Values of A_x and A_y versus angle ψ

Clearance s between the ring ends in the free condition is equal to the value of $2Y$ for $\psi = 180°$, i.e. 9.42 (see Fig. 295), multiplied by the appropriate proportionality factor:

$$s=9.42\,\frac{\sigma_{max}}{4E}\,\frac{D_0^2}{b}$$

Design of Piston Rings. Piston rings of rectangular cross section are used most widely (Fig. 296, *I*). The edges of the ring inner periphery are provided with chamfers of (0.2 to 0.5) $\times$ 45° in order to prevent location of the ring on the bottom fillets of the piston-ring grooves and simplify the assembly of rings with the piston. Labyrinth grooves are made on the outside periphery of large-diameter rings (Fig. 296, *II*).

For increased pressure on the cylinder walls, the ring outside surface can be provided with circular recesses (Fig. 296, *III*, *IV*);

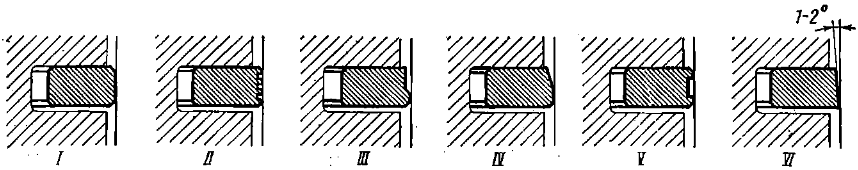


Fig. 296. Piston-ring cross-section profiles

this design, however, results in a reduced radial sealing effect because the fluid pressure on the recessed portion of the outer periphery counteracts the pressure on the ring inner surface.

The above condition is used to obtain uniform distribution of load between individual rings of the set. Owing to the recesses, the first rings (nearest to the pressure chamber of the cylinder) exert lower pressure on the cylinder walls and, thereby, the other rings become loaded additionally. This method is applied to hydraulic cylinders, piston-compressor cylinders, etc. It also proves helpful for vacuum cylinders, where vacuum causes the rings to depart from the bore surface and where, therefore, the corresponding decrease in the radial sealing effect must be compensated for.

In internal combustion engines, the recesses are not made in the first compression rings since this may result in the deposition of carbon on the rings due to the entry of combustion products in the clearance between the ring and the cylinder walls. The recesses are only provided on the last rings, which are exposed to a pressure considerably weakened by a throttling effect of the preceding rings and in which sealing is caused not so much by the radial action of the pressure as by the rings' inherent resilience. The recesses similar to those shown in Fig. 296, *V* have little influence on the radial sealing effect.

In order that piston rings may wear in to the cylinder bore more rapidly, the ring outside surface is made tapered (Fig. 296, *VI*), with a narrow cylindrical land (0.3-0.5 mm) left near the face. This calls for machining every ring to taper separately.

The tapered rings can be produced more efficiently by a method in which they are clamped in a group between massive tapered discs and ground on the outside diameter (Fig. 297, *I* and *II*, respect-

ively). After being removed from the workholding device, the rings straighten out, and their periphery takes a slightly tapered form. This method allows piston rings with a taper of 20′ to 30′ of arc to be produced. Such a taper is difficult to see with the unaided eye. To prevent faulty assembly, the taper direction should be indicated with a mark on the ring face.

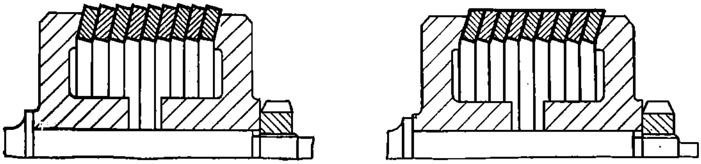
Another method of obtaining a tapered working surface on a piston ring is based on the tendency of asymmetrical sections to twist under the action of bending forces. A recess or a tapered portion which displaces the principal axis of inertia of the section with respect to the direction of bending force is made on the ring inner surface (Fig. 298, *I*, *II*). When inserted into the cylinder, such rings are twisted by the compressive action of the cylinder walls; as a result, the outside surface of the ring takes a tapered form (Fig. 299). The taper is variable around the ring's circumference; it is maximum at the joint. The friction of the ring's edge against the cylinder walls during the downward movement of the piston also makes for the twisting of the ring. Owing to simple manufacture, the twisting rings have found wide application.

Trapezoidal-section piston rings (Fig. 300, *I*, *II*) are employed in cylinders operating at high temperatures (in internal-combustion engines, high-pressure piston compressors, etc.), where there is a hazard of the rings' carbonization due to decomposition of oil at high temperatures. A tapered form of the ring is advantageous for the extrusion of the deposits out of the piston-ring grooves on each reversal of piston movement, whereby the rings retain their freedom to move in the grooves. Moreover, the trapezoidal rings exert elevated pressure on the cylinder walls because of a wedge-like action of the groove walls during piston movement.

Fig. 301, *I*, *II* shows some section profiles of twisting trapezoidal rings.

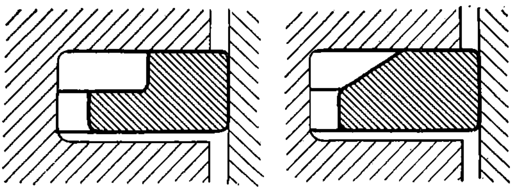
Oil-Scraper Rings. In gas cylinders, the entry of lubricating oil into the working chamber has to be prevented. The objective is achieved by the use of oil-scraper rings placed before the common piston rings (called here gas rings) along the direction of piston movement.

The oil-scraper rings remove oil from the cylinder walls, thus providing against its coming to the gas rings and, subsequently, penetration into the working chamber. The oil scraper rings of all types have the following features in common: (1) they exert an elevated pressure on the cylinder walls owing to reduced frictional surfaces; (2) they are provided with cavities to collect the scraped oil; (3) the oil being removed is drained through passages that connect the piston-ring grooves with the inside space of the piston; and (4) they are installed in the piston-ring grooves with a large-clearance fit.



I II

Fig. 297. Machining tapered piston rings in a group



I II

Fig. 298. Cross-sectional profiles for twisting rings

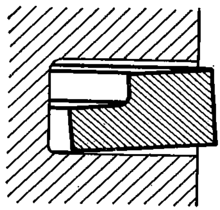
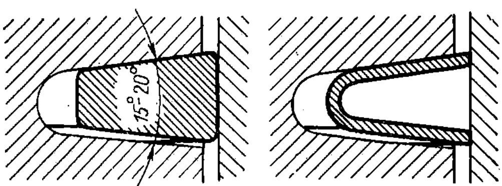
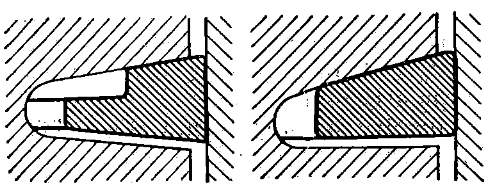


Fig. 299. Asymmetrical-section piston ring twisted by radial pressure



I II

Fig. 300. Trapezoidal-section rings



I II

Fig. 301. Twisting trapezoidal rings

The rings shown in Fig. 302, *I*, *II* have the form of a scraper. The oil scraped from the cylinder-bore surface is removed through the face clearance in the piston-ring groove and radial passages in the piston walls.

An additional oil-removing cavity communicating with the back surface of the ring through radial channels is provided in the design according to Fig. 302, *III*. In the ring shown in Fig. 302,

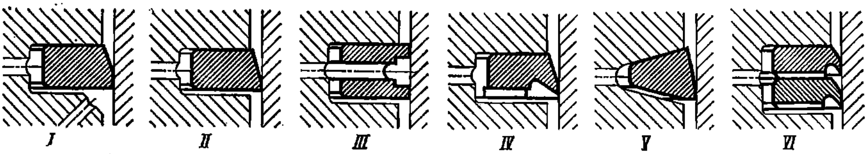


Fig. 302. Oil-scraper rings

IV, oil is removed from below the scraper through slots cut in the ring's face. Fig. 302, *V* shows an oil scraper ring of a trapezoidal cross section. A double arrangement of oil-scraper rings (Fig. 302, *VI*) is employed in heavy-duty cylinders.

Piston-Ring Joints. A straight slit (Fig. 303, *I*) is the simplest type of joint; its disadvantage is that the ring ends exert an elevated

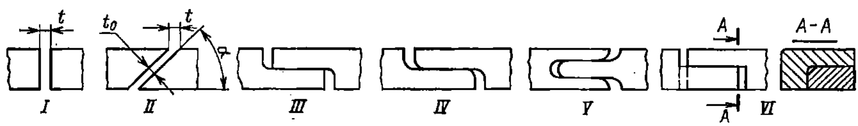


Fig. 303. Types of piston-ring joints

pressure on the cylinder walls and wear out the bore surface. The escape of fluid through such a joint is relatively high.

Slanting-slit joints (Fig. 303, *II*), wherein the pressure on the cylinder walls is more uniform owing to the gradually narrowing ends, are better; these joints offer a higher sealing effect because of a longer way of the fluid through the joint. Moreover, as the slit width t is specified in a transverse direction, wherein the ring ends close, normal clearance t_0 that controls the flow of the escaping fluid is here narrower than in a straight slit. Clearance t_0 is determined by the formula $t_0 = t \sin \alpha$ (where α is the slit-slanting angle). For the most frequently used angle, $\alpha = 45°$, $t_0 = t \sin 45° \approx 0.7\,t$.

The sealing capacity of step joints (Fig. 303, *III*-*V*), wherein the clearance is essentially equal to zero, is still higher. These joints, however, are more difficult to make; in addition, with narrow rings, the tongs come out too thin and break off easily. For increased strength, it is good practice to join the tongs to the ring body with fillets (Fig. 303, *IV*, *V*).

Figure 303, *VI* illustrates a tight step joint with the steps disposed in mutually perpendicular planes. The escape of gas through this joint is substantially lower than through the foregoing types of joints; this construction, however, is much more difficult to produce.

Locking of Piston Rings. As piston rings are free to move in their grooves in the piston, there is a hazard of alignment of the ring joints, which may result in increased leakage. To prevent this condition, the rings are fixed against rotation with radial pins

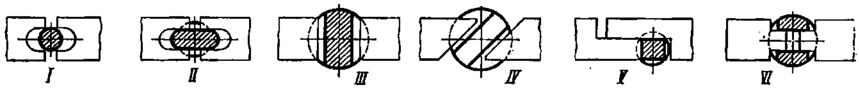


Fig. 304. Fixing methods for piston rings

secured in the piston body and introduced into the ring joint. The adjacent rings are so located as to have their joints disposed in a diametrically opposite relation.

The piston-ring locking methods are shown in Fig. 304, *I-VI*. The drawback to the fixed rings is that they wear out the cylinder bore surface unevenly (because of nonuniform pressure around its circumference) and thus render the bore out-of-round. With movable, unlocked rings, nonuniform wear is made up for by uncontrollable circular displacements of the rings in their grooves during operation. The circular displacements with the slanting-slit rings are fairly regular because the shifting forces that arise in the ring joint during reciprocation of the piston tend to turn the ring in the groove.

Fixing of piston rings is mandatory if the cylinder walls have recesses, ducts, or windows (e.g. scavenging ports in a two-cycle internal combustion engine), which are passed across by the rings as the piston reciprocates. If the ring joint crosses an opening accidentally, the ring can break.

Materials and Manufacture. Piston rings are produced most commonly from quality pearlitic cast iron; this material exhibits high wear-resistance and antifriction properties which are due to presence of flaky graphite in its structure.

Mechanical properties of cast irons used for making piston rings normally vary within the following limits (the upper limit applies to alloyed cast irons): $E = 11{,}000$ to $13{,}000$ kgf/mm²; $\sigma_t = 30$ to 50 kgf/mm²; $\sigma_b = 40$ to 60 kgf/mm²; $\sigma_{0.2} = 20$ to 30 kgf/mm²; BHN 100 to 120; and $\delta = 0.2$ to 0.6%.

These properties are practically unchangeable up to 450 °C.

After being rough machined, cast-iron piston rings are subjected to artificial (ageing at 500-550 °C) or natural ageing.

Piston rings designed to operate with profuse lubrication, are produced from spring steel, which is hardened and tempered at 350 to 500 °C. Steel piston rings call for higher surface hardness of the cylinder bore.

In some cases, piston rings are made from wrought bronze, grades БрАЖН or БрАЖМЦ, and for vital applications, from berillium bronze, grade БрБ2.

Elliptical piston rings, providing uniform pressure, are manufactured by one of the following methods: (1) casting to shape (for cast-iron rings); (2) machining to a template; (3) deforming and subsequently heat treating the blank to retain the shape (the thermal method); (4) rolling on the inside periphery under variable pressure.

Cast-iron piston rings intended for critical applications are cast into permanent moulds to obtain blanks with minimal allowances for subsequent machining.

Piston-ring blanks are machined to a template by turning or milling. The joint slit is then cut, the ends are closed up, and, in this condition, the outside and inside diameters are ground.

When produced by the thermal method, round blanks made with small stock allowance are set on a mandrel whose form corresponds to the designed profile. The obtained ring profile is rendered permanent by heating the blanks to 600-650 °C. The blanks are then transferred to finishing operations, where they are processed with the closed ends.

To be made by rolling, the ring blanks are loaded into annular grooves of a special rolling apparatus. The ring's inside periphery is processed with a roll installed in the apparatus eccentrically in such a way as to exert the maximum pressure on the side opposite to the joint. With a properly selected eccentricity, the rolled ring removed from the apparatus will take the form close to that specified. The subsequent operations are grinding the faces and outside diameter with the closed ends.

The rolling results in strain hardening: in inner grains of the material compressive stresses arise that counteract the tensile stresses caused by straining the ring as it is fitted into the piston groove. Thereby, the ring width may be safely increased and a gain in pressure obtained.

After finish-machining operations, the ring is lapped to a master cylinder. The accuracy of the matching is checked by sighting the clearance between the outside contour of the ring and the cylinder-bore surface. The permissible values of clearance are specified depending on service conditions. For precision rings, the clearance should not exceed 0.01 mm.

The rings to be used in critical applications are checked for uniform radial pressure by assessing the pressure polar diagrams produced with the aid of special measuring devices.

Coatings. For increased wear resistance and service life, the working surface of piston rings is subjected to chromium plating. A chromium-plated surface features an extremely high hardness (900 to 1,000 HV) and heat resistance, low coefficient of friction, and good anti-galling properties.

In hard chromium plating, chromium is deposited in a solid layer of 0.15 to 0.25 mm thick on small rings, and up to 0.5 mm thick on large rings.

The rings plated with a thin layer undergo no subsequent machining before coming into service, and the thick-layer rings are ground to correct nonuniformity of plating.

Hard chromium plating has the following shortcomings:

(1) the ring's wearing-in process is rather long because the chromium layer exhibits high hardness and poor oil-wetting properties;

(2) the rings call for cylinders produced to higher accuracy and have to mate with them without a clearance.

These shortcomings are obviated to a considerable extent with *porous plating.* Chromium is first deposited in a solid layer; it is then made porous (by changing the direction of electric current at the final phase of the plating process) to a depth of about 0.25 the layer thickness.

A porous surface holds oil properly. In the process of wearing-in, a porous layer wears away rather quickly (particularly, on the portions subjected to elevated pressures), exposing the bottom layer of solid hard chromium. The presence of oil in the porous layer prevents galling during the wear-in process.

The wear resistance of porous-chromium plated rings largely depends on the structure of the porous layer, which determines the character of wearing-in. A grid-like porous pattern with a size of pores from 0.05 to 0.1 mm^2 offers the best results. With a correctly effected wear-in process, the chromium-plated rings will have 15 to 20 times the wear resistance of conventional cast-iron rings.

The material of chromium-plated rings is not so critical as that of uncoated rings. For this reason, chromium plated rings may be produced from strong inoculated cast-iron containing globular graphite, or from steel.

The cylinder bore can also be chromium-plated. This is a costlier process than the piston-ring plating because the plated bore surface has to be carefully finished. The process, however, opens the possibility of making cylinders from aluminium alloys which exhibit high thermal conductivity, vitally important for cylinders working at elevated temperatures.

Other processes employed to improve the wear resistance of piston rings are as follows.

Oxidizing. The rings are kept in a gas-oxide and water-steam atmosphere at 500 to 550 °C to obtain the surface coated with a thin layer (0.01 mm) of magnetic ferrosoferric oxide Fe_3O_4.

Phosphatizing. The rings are held in a hot water solution of phosphoric acid saturated with phosphates of Fe, Mn, and Zn. A porous crystalline phosphate layer readily absorbing lubricants forms on the ring surface.

Diffusion siliconizing. The rings are covered with powdered silicon carbide, SiC, and held at a temperature of about 1,000 °C. The surface layer is saturated with silicon, whereby its wear resistance is improved.

Chromizing. The outer layer is saturated with chromium by holding the rings in melted chromium chloride, $CrCl_2$, or in an atmosphere of gaseous chromium chlorides at about 1,000 °C.

Alitizing. The rings are kept in a mixture of powdered aluminium and aluminium oxide, Al_2O_3, at a temperature of about 1,000 °C; the solid-solution crystals of aluminium in α-iron form in the outer layer, and the surface becomes coated with a thin wear-resistant film of aluminium oxide.

Sulphurizing. The rings are kept in a hot solution of sodium hydroxide, NaOH, with some addition of sulphur, or in a melt of sodium cyanide, NaCN, and sodium sulphate, Na_2SO_4. The sulphurized layer has exceptionally good resistance to wear and thermal seizure.

For faster wearing-in, piston rings are subjected to galvanic tinning, cadmium plating, or copper plating. Tinning gives the best results. Galvanic plating is effected in a bath of tin-acid sodium salt at 75 °C. The tin layer produced is 0.005 to 0.01 mm thick.

Piston rings functioning at moderate temperatures are coated with a thin layer of epoxy resin, fluoro-plastic, etc., with a graphite or metal-powder additive.

Chapter 3

Sealing Stationary Joints

3.1. Gaskets

Flat interfaces between parts are packed most extensively with gaskets made from elastic materials. As a rule, installed with gaskets are covers of oil-containing reservoirs functioning under pressure or vacuum, pipe flanges, etc. Where no strict mutual-position requirements are specified, soft gaskets may be used to pack joint faces in mechanical-transmission housings.

A gasket material is selected depending on operating conditions, magnitude of pressure, temperature, etc. To seal general-purpose non-critical joints, e.g. the covers of oil reservoirs, use is made most often of gasket paper of 0.05 to 0.15 mm thick, paper impregnated with bakelite or other synthetic resins, gasket cardboard of 0.5 to 1.5 mm thick, etc. Gaskets of synthetic materials, such as polyvinylchloride or polytrifluoroethylene, offer the best packing properties.

For high-temperature applications, asbestos-base gasket materials are employed (asbestos paper, asbestos cardboard, etc.). Steam pipes are most frequently sealed with paronite, which is a composition of asbestos and natural or synthetic rubber. Paronite stands up to 450 °C. Also used at high temperatures are gaskets of ductile metals: aluminium sheet, aluminium and copper foils, etc. Such gaskets require an elevated tightening pressure.

Where dimensional adjustments between components of a unit are needed along with sealing, use is made of shims, i.e. sets of brass- or copper-foil gaskets about 0.05 mm thick. An instance of application of shims is double arrangements of tapered or self-aligning antifriction bearings, where shims are used for packing as well as for adjusting the amount of tension.

For better function, gaskets are coated with sealing greases, or sealants. Paper and cardboard gaskets are treated with boiled oil, shellac, raw bakelite, soluble glass, minium, white, etc.

Good sealing properties are provided by a grease comprising 35 per cent shellac, 55 per cent alcohol, 6 per cent flake graphite, 3 per cent castor oil, and 1 per cent colouring agent (ochre).

Sealants of various compositions, substantially on the basis of natural or synthetic rubber with appropriate solvents, are exten-

sively used. For high-temperature applications, use is made of heat-resistant sealants: ethylsilicate, siloxane enamels, etc.

Soft-material gaskets are not re-usable and should be replaced after each disassembly of a joint.

Rarely dismantled joints are sealed with packing materials that flatten out in the interface by compression: oil-impregnated cotton

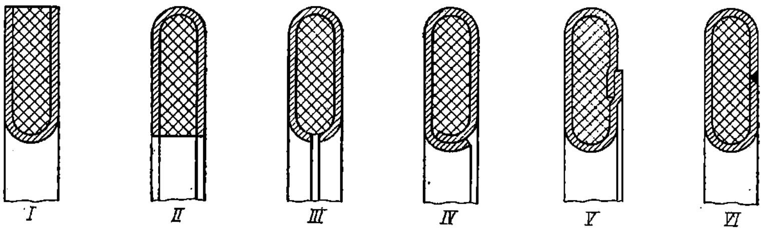


Fig. 305. Reinforced gaskets

yarn, rubber yarn and cord, greased asbestos cord, wire of lead, aluminium, or annealed copper, etc. The last two kinds of packing materials (asbestos and wire) are applicable for joints operating at high temperatures.

Soft-material gaskets are used for joints tightened with bolts, studs, etc., where the gasket is in compression only. The gaskets subject to shear (in threaded connections, plugs, etc.) are produced from stronger and stiffer materials, such as fibre, lead, or annealed copper.

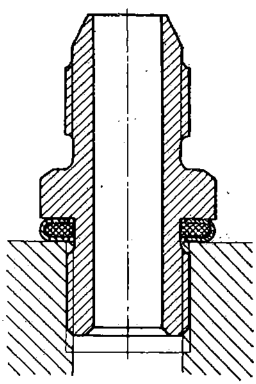


Fig. 306. Nipple packed with reinforced gasket

Reinforced gaskets consisting of an elastic gasket material (rubber, plastic, asbestos, etc.) encased in a shell of soft metal (copper or brass) are also applicable. Some varieties of such gasket units are shown in Fig. 305, *I-VI*.

Figure 306 illustrates the use of a reinforced gasket for packing a nipple. The reinforced gaskets are re-usable.

Placing of Gaskets. Gaskets should be fixed with respect to the surface being packed and tightened over the whole bearing area.

Typical gasket-placing defects are illustrated in Fig. 307. In the joint according to Fig. 307, *I*, the gasket is not fixed radially; it can be displaced or extruded in tightening. Centring a gasket on the unthreaded portion of a cap screw (Fig. 307, *II*) gives no desired result because it is done by feel, after the gasket and the washer are put in place; a substantial

area of the gasket, disposed over the hole, is unsupported and thus not compressed by the surfaces being packed.

A correct design is presented in Fig. 307, *III*. Here, the gasket is centred on the cover spigot and thus protected against extrusion inwards.

In joints assembled with studs (Fig. 308, *I*), the plain portions of the studs may be used for centring the gasket; the latter is fitted

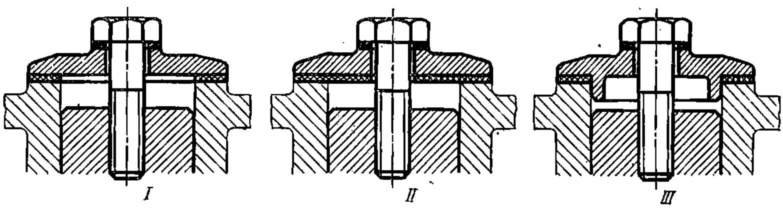


Fig. 307. Gasket placing methods
I, *II*— incorrect; *III*—correct

over the studs and secured from above with the part being fastened, e.g. a cover. Where fastening is effected with cap screws, it is advisable to provide the cover with a centring spigot (Fig. 308, *II*); here, the gasket is put on the cover spigot, aligned so that its holes coincide with the clearance holes in the cover, and the latter is fastened together with the gasket to the housing.

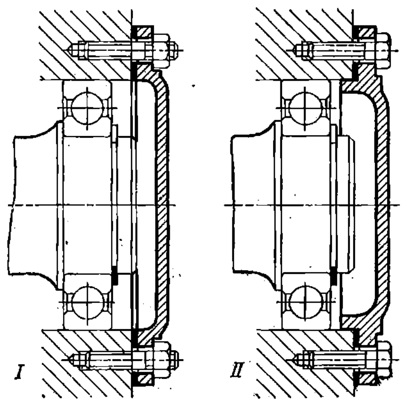


Fig. 308. Gaskets fixed with bearing covers

It should be taken into account that gaskets, especially thick ones, become deformed in tightening. If a gasket is so dimensioned as to exactly correspond to the nominal delimiting dimensions of the interface (Fig. 309, *I*), it will be extruded by the tightening forces outwards, which will deteriorate the exterior of the assembly, and inwards, which can perceptibly narrow the effective section of passages and conduits (Fig. 309, *II*).

Here, the outer dimension of the gasket should be taken to be somewhat smaller (by 0.5-1 mm), and the inner dimension, somewhat larger than the nominal (Fig. 309, *III*) so that the contour of the compressed gasket coincides with that of the surface being sealed.

The roughness of joint faces should be held to within 1.6 μm Ra (Fig. 310) if adequate tightness is to be obtained.

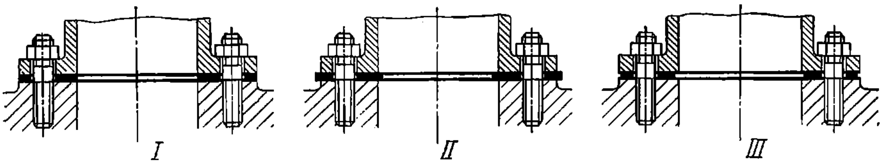


Fig. 309. Radial dimensioning of gaskets

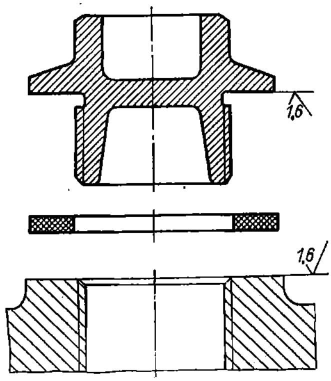


Fig. 310. Surface roughness of joint faces

3.2. Sealing Rigid Joints

The use of soft-material gaskets always causes some alteration of space between the surfaces being sealed. In many engineering products, (e.g. in multi-section built-up housings containing plain or antifriction bearings) it is required to seal metal-to-metal joints and, at the same time, to secure an accurate mutual position of mating parts.

Rigid joints can be packed by various methods. Permanent or rarely detachable joints are packed with sealing compositions: bakelite, white, minium, soluble glass, etc.

A wide variety of sealants designed for use in different joints are available. Among these are:

(1) grade У-30М based on thiocol rubber; it is resistant to oil, benzine and water, and is highly gas-tight; the working-temperature range is from —50 to +130 °C; adhesion to metals is low;

(2) grade ВТУР based on thiocol with dynzocyanate; it is oil-, benzine- and water-resistant; the working-temperature range is from —50 to +130 °C; adhesion to metals is high;

(3) grade ВГХ-180 phenol-formaldehyde resin with natural rubber; it is oil- and water-resistant; the working temperature range is from —50 to +130 °C; adhesion to metals is high; swells under the action of benzine and kerosene;

(4) grade 5Ф-13, fluoro-elastomer with epoxy resin ЭД-6; it is benzine-, oil-, and water-resistant; the working temperature range is from —50 to +200 °C; adhesion to metals is low; and

(5) grades ВИКСИНТ У-1-18 and ВМТ-1, based on silicones; they are resistant to oil and water; temperature resistance is up to 300 °C; swell on exposure to benzine and kerosene; adhesion to metals is low.

Sealants come as greases or lacquers. They are applied to the surfaces being packed by pouring, with a brush, or with a spatula. The coat makes a stable film in about five or six days.

Silicone enamels (silicones with metal-powder fillers—Al, Zn), which withstand temperatures up to 800 °C, are suited to joints operating at particularly high temperatures.

In tightening, a surplus of the sealant is extruded out of the interface, and a thin film with a thickness of several microns or hundredths of a millimetre remains only; this film has practically no effect on the accuracy of mutual position of the parts joined.

The units put together with the application of sealants are difficult to disassemble, especially after operation at high temperatures. Suitable puller devices should be provided to handle such units.

Precision metal-to-metal joints are made tight by fine finishing the surfaces sealed, e.g. by lapping or scraping.

Subjected to *lapping* are parting planes previously finish shaped with broad-nose tools, finish milled or ground. Lapping is done on plate laps of cast iron or special glass (pyrex) with accurately processed surfaces. The article is pressed against the lap, to which a small-amplitude circular movement is imparted.

Lapping abrasives are varied. Used most frequently are powders of glass, silicon carbide, corundum (crystalline aluminium oxide), boron carbide, and diamond (for hard metals). Machine oil, kerosene, and fatty acids are applicable as lubricants.

Lapping is first effected by means of grinding powders with a grain size not over 100 μm, and then by micro-powders. The process is completed with ГОИ-paste consisting essentially of chromium oxide with binding and lubricating additives (stearine, kerosene, oleic acid, etc.). In some cases, the surfaces to be joined are lapped directly against each other.

Lapping is a costly and labour-intensive process; for this reason it is only applied to vital joints. The process has recently been mechanized. In some instances, it can be replaced by more productive finish-shaping and milling operations.

Scraping is usually effected in the following sequence. First, one parting plane is scraped to a surface plate so as to obtain from two to five marking spots over 1 cm². A thin layer of marking blue is then applied to the scraped surface, and the mating part is put on it. With slight circular movements, the marking-blue layer is transferred to the mating surface, and the resultant marking spots are scraped off. The operation is performed repeatedly until the desired accuracy of mating is achieved. Scraping is a very labour-consuming process and is seldom used in quantity production.

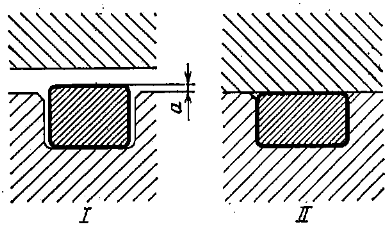


Fig. 311. Metal-to-metal joint packed with elastic gasket

I and *II*—gasket before and after assembly

The lapped and scraped surfaces are covered in assembly with a thin layer of a sealing paste. The latter is made most often from a low-consistent paint diluted with boiled oil (lead white, minium, ochre, etc.), or iron powder mixed with oil. Also used is colloidal-graphite suspension in oil. In some cases, mating surfaces are rubbed with dry silver graphite.

High stiffness of flanges and closely arranged screw fasteners are prerequisites of reliable sealing.

Another method of sealing rigid joints consists in placing elastic rectangular or round gaskets flush with the joint planes. The gaskets are let into grooves cut around the whole periphery of the joint. In the free condition, the gasket projects over the joint plane by a strictly specified amount a (Fig. 311, *I*), dependent on the gasket material and the desired packing pressure. In tightening, the joint faces are brought into contact, upon which the gasket material becomes deformed elastically or permanently, thus packing the joint (Fig. 311, *II*).

For better tightness, the surfaces being packed are provided with fine serrations which the gasket material enters under pressure (Fig. 312). Alternatively, fine teeth may be made on a gasket (Fig. 313). When such a gasket is compressed, the teeth are flattened, and a row of pockets acting like a labyrinth seal are formed.

The groove cross section should exceed that of the gasket so as not to hinder the gasket's deformation.

The gasket material is selected depending on the joint's operating conditions. Rubber or plastics are used for applications under average service conditions, and ductile metals (lead, aluminium, annealed copper, etc.), for high-temperature applications. Cadmium-plated copper also provides a good sealing capability.

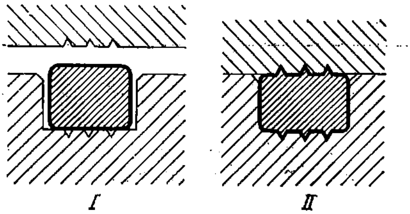


Fig. 312. Metal-to-metal joint provided with fine serrations

I and *II*—gasket before and after assembly

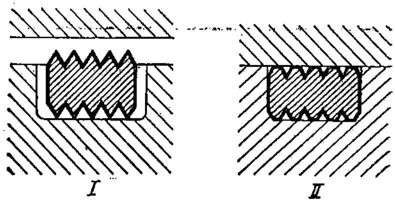


Fig. 313. Gasket with fine teeth to pack metal-to-metal joint

I and *II*—gasket before and after assembly

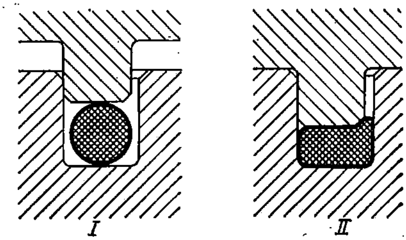


Fig. 314. Metal-to-metal round-flange joint

I and *II*—packing ring before and after assembly

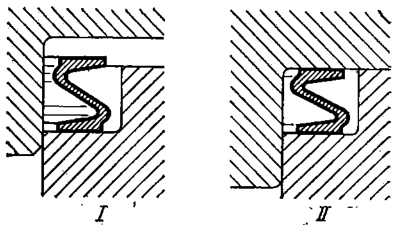


Fig. 315. Joint packed with bellows-type flexible rings

I and *II*—ring before and after assembly

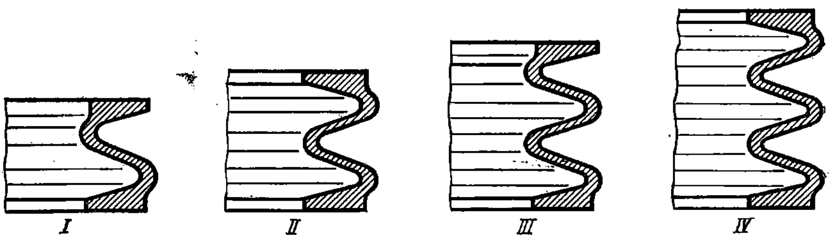


Fig. 316. Bellows-type flexible rings

I—Z-type; *II*—with a single convolution; *III*—with one and a half convolutions; *IV*—with two convolutions

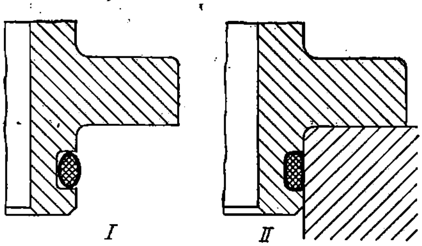


Fig. 317. Metal-to-metal joint packed with elastic cord

I and *II*—packing before and after assembly

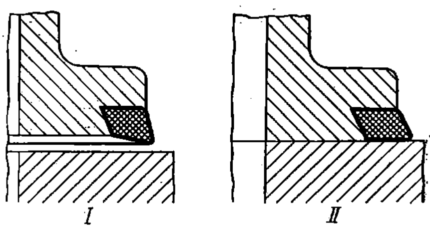


Fig. 318. Metal-to-metal joints packed with peripheral annular gaskets

I and *II*—gasket before and after assembly

Fig. 314 shows an elastic gasket placed in a confined space formed by a groove in one mating surface and a projection on the other mating surface. This design is mainly used for round flanges wherein the grooves and projections can be machined by turning to the required accuracy.

Round flanges are also packed with flexible metal rings (Fig. 315), mostly of a Z-cross section, which are similar to bellows. Shapes

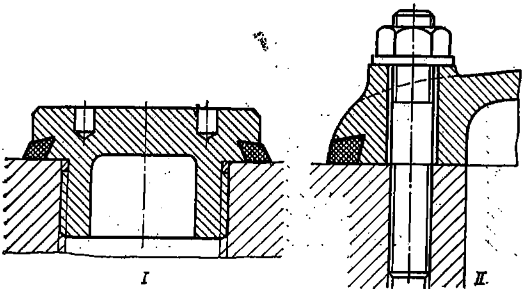


Fig. 319. Packing with peripheral gaskets

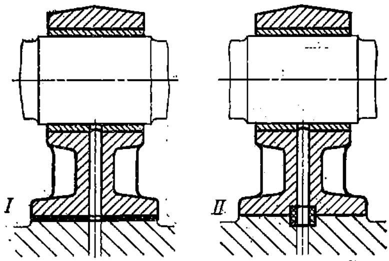


Fig. 320. Packing oil hole in plain-bearing pillow block
I—incorrect packing; *II*—correct packing

of bellows-type rings are shown in Fig. 316 in order of increasing compliance.

Round flanges with centring collars are packed with cord of elastic materials (rubber, plastics), which are let into suitable grooves cut in the collar (Fig. 317). With this arrangement, a straight metal-to-metal contact is ensured. This design is only used for medium-temperature applications.

Fig. 318 shows an annular gasket placed in an open peripheral groove. The advantage of the design is that the gasket protects the interface against corrosion and other environmental influences. Joints with peripheral gaskets are also shown in Fig. 319, *I*, *II*.

In some metal-to-metal joints, round openings and passages serving for the supply of lubricating oil, drainage of coolant, etc. need to be sealed.

Fig. 320 illustrates an instance of packing the oil hole in a plain bearing. A soft gasket (Fig. 320, *I*) is here unsuitable because the shaft position with respect to the adjacent parts will be upset in tightening the pillow block: e.g., if the shaft is driven by a gear transmission, the tightening will upset the correct engagement of the gear wheels. To avoid this condition, packing inserts are used (Fig. 320, *II*).

Fig. 321, *I-III* displays the inserts made from elastic materials (rubber, plastics, etc.). The packing effect is achieved by axial

(Fig. 321, *I*, *II*) or radial (Fig. 321, *III*) compression of inserts. In some cases, metal bushings in combination with elastic packing

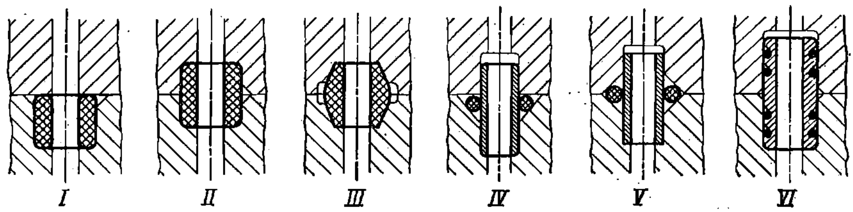


Fig. 321. Packing inserts

elements are employed (Fig. 321, *IV-VI*). The bushings can also be used as dowel pins.

3.3. Flange Packings

Fig. 322 shows methods for packing cylindrical flanges, e.g., flanged housing covers.

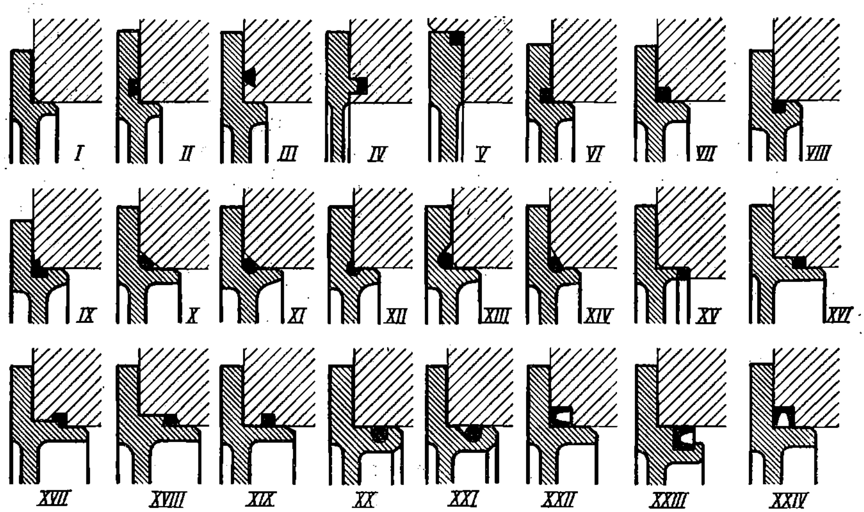


Fig. 322. Flange packings

The simplest soft-material gasket packing is shown in Fig. 322, *I*. The rest of the packings in Fig. 322 relate to those used in metal-to-metal joints.

Fig. 322, *II-VI* presents instances of packing with cord of elastic materials (rubber, plastics, etc.) mounted in flange or housing

face grooves. Such face packings tend to increase the radial spacing of cap-screw fasteners, and, consequently, increase the flange diameter; moreover, a groove in its body weakens the flange. Better in this respect are hole-edge packings (Fig. 322, *VII-XIV*). Designs wherein the packing cord is let into a groove in the flange collar to form a whole with it during assembly are the most convenient (Fig. 322, *VIII*, *IX*, *XI*, *XII*, *XIV*).

The packings in Fig. 322, *X*, *XI*, *XIV* are designed to operate at a high fluid pressure in the chamber packed and are similar in principle to lip packings: the fluid forces the cord into the groove's narrowing space and thus enhances the pressure of the cord against the surfaces being packed.

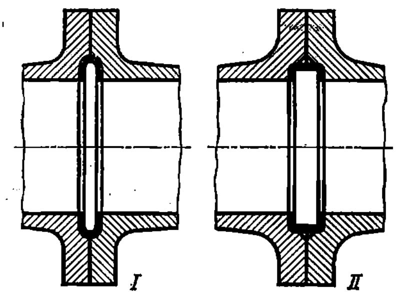


Fig. 323. Lip packings in pipe-flange joints

Fig. 322, *XV-XVIII* shows packings wherein the cord is mounted in an annular space between the flange collar and the housing and is compressed in an axial direction. In the design shown in Fig. 322, *XV*, the gasket is liable to be extruded out of the groove; therefore, stiff gaskets have to be used here.

Fig. 322, *XIX-XXI* presents radial packings wherein the cord is crowded into a circular groove made in the housing or in the flange collar; the sealing effect is due to radial deformation of the cord during mounting the cover. Designs wherein the cord is fitted into the collar groove are the most suitable for assembly. An inclined groove, as in Fig. 322, *XXI*, causes the packing cord to function similarly to a lip packing. Fig. 322, *XXII-XXIV* presents straight lip packings applicable on large flanged covers.

Fig. 323, *I*, *II* shows lip packings in pipe flanges.

3.4. Sealing Screw-Threaded Assemblies

Fig. 324 illustrates methods for packing sleeve nuts with annular-groove gaskets and cords. As the gaskets are subjected to shearing forces when the nut is tightened, the gasket material must be fairly stiff.

Fig. 324, *I-VI* shows edge-type cord packings placed in a circular groove cut in the body of a sleeve nut; Fig. 324, *VII-XI* shows cord packings compressed endwise in a restricted annular space between the nut and the housing; and Fig. 324, *XII-XV* shows radial packings using cord let into a circular groove in the nut or housing.

Sealing Screwed Components. The simplest method of packing screwed fitting-type components (nipples, plugs, etc.) is to apply a sealing grease to their threaded portions. This, however, leads to adhesion of the grease to the thread after some period of service, which will make unscrewing difficult.

Used sometimes in repair practice the coiling of hemp fibres greased with oil-diluted minimum onto the threads adjacent to the part end is objectionable.

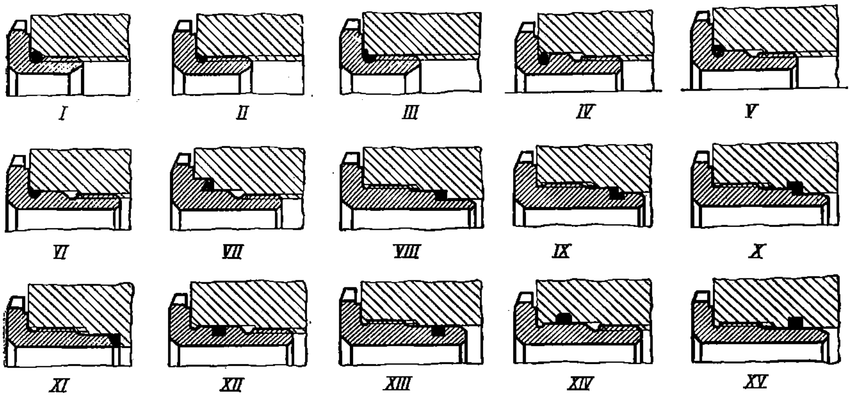


Fig. 324. Sleeve nuts packed with cord

Fig. 325 illustrates methods for packing nipples with elastic gaskets. In the design shown in Fig. 325, *I*, the gasket is exposed to action of the tightening pressure to the fullest extent. To prevent the gasket from being crushed, it must be made from a hard or medium-hard material, or be of a reinforced type; alternatively, the tightening pressure should be restricted.

The gasket in Fig. 325, *II* is placed in an annular space formed by a recess in the housing. The gasket material can only be forced into the thread, which improves sealing conditions.

In the designs according to Fig. 325, *III*-*V*, the sealing effect is achieved through deformation of the gasket as the nipple is screwed in to bear with its shoulder on the housing face; here, the gasket deformation is determined by the difference in height between the gasket and the packing groove.

The nipples shown in Fig. 325, *VI*, *VII* are packed on their screwed-in ends. As in the previous designs, tightening is done until the nipple shoulder comes up against the housing. The gasket in Fig. 325, *VII* is mounted in a confined space and, in contrast to that in Fig. 325, *VI*, cannot be extruded in tightening. The nipple can be tightened to bear up either against the gasket or the housing;

in the latter case, the volume of the packing groove must exceed that of the gasket. The packing pressure is determined by the difference between the heights of the gasket and the groove (with the nipple being screwed in to the full).

The gasket can be placed in a radial groove cut in the nipple shank (Fig. 325, *VIII*). When the nipple is screwed in, the gasket

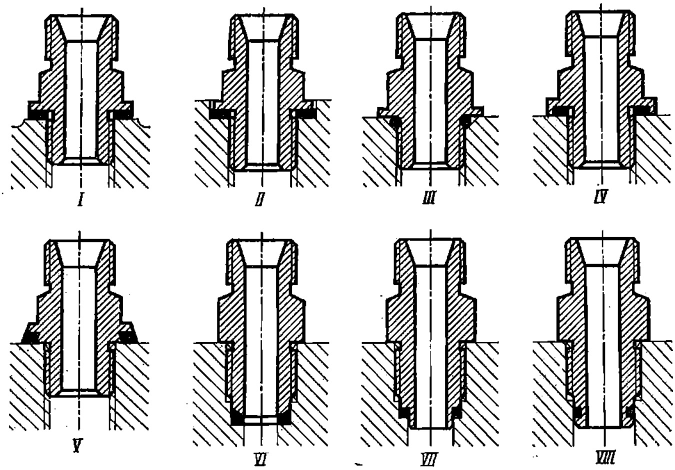


Fig. 325. Nipples packed with elastic gaskets

moves with it relative to the housing. The packing pressure depends here on the amount by which the gasket projects from the groove in the free condition.

Fig. 326 presents instances of packing nipples without gaskets or with the use of metallic packing elements. A taper screw thread (Fig. 326, *I*) ensures complete tightness, especially if the housing is made from a ductile metal. The rest of the designs shown in Fig. 326 are based on permanent deformation of the housing or nipple materials. These designs may be used for rarely dismantled or permanent joints.

Some methods for packing with sharp annular edges are illustrated in Fig. 326, *II*, *III*. The edge is made on the harder-material part (the housing in Fig. 326, *II*, and the nipple in Fig. 326, *III*), and during the process of screwing-in, it penetrates the softer material, thus securing tightness. Packing may be effected similarly with the use of separate sharp-edged rings (Fig. 326, *IV*, *V*). In this case, the materials of the nipple and the housing must be softer than the ring material.

The packing methods that use permanent deformation of the housing thread are illustrated in Fig. 326, *VI-VIII*. Incomplete

thread on the nipple end screwed into the housing crushes the entering threads in the latter and thereby provides tightness in the joint (Fig. 326, *VI*). The shank of the nipple shown in Fig. 326, *VII* is furnished with a conical portion which, as the nipple is screwed in, upsets the entering threads in the hole; in this way, the joint becomes fixed rigidly as well as packed. In the design in Fig. 326,

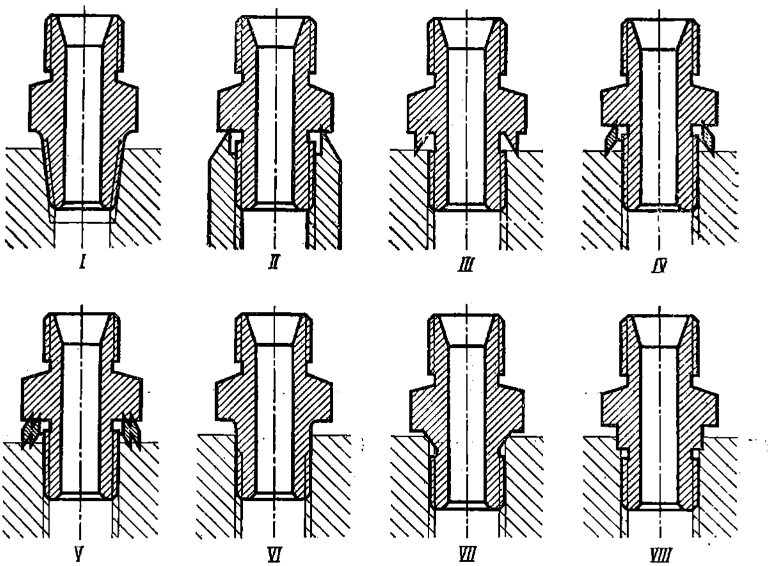


Fig. 326. Nipples packed without gaskets or with metal packing elements

VIII, a cylindrical collar on the nipple shank serves the same function. The joints displayed in Fig. 326, *VII*, *VIII* are permanent.

Tight Screw-Threaded Assemblies. Fig. 327 shows methods of sealing large-diameter tight-screwed assemblies designed to operate at high internal pressures and temperatures. Such joints make use of interference-fit screw threads; before assembly, the outer screwed part is heated or the inner, cooled.

The thread is produced with high accuracy by milling or grinding and, prior to assembly, is lubricated with sealing greases. For better heat conductivity, metal fillers (aluminium, bronze, or zinc powders) are added to the greases.

Tightness is also secured by the following additional measures: the parts joined abut on their face portions directly (Fig. 327, *I*) or through gaskets (Fig. 327, *II*, *III*) made from ductile metals (lead, copper, or aluminium); one of the parts may be provided with a sharp circular edge that penetrates the face of the mating part (Fig. 327, *IV*-*VI*); mating parts are fitted together on accu-

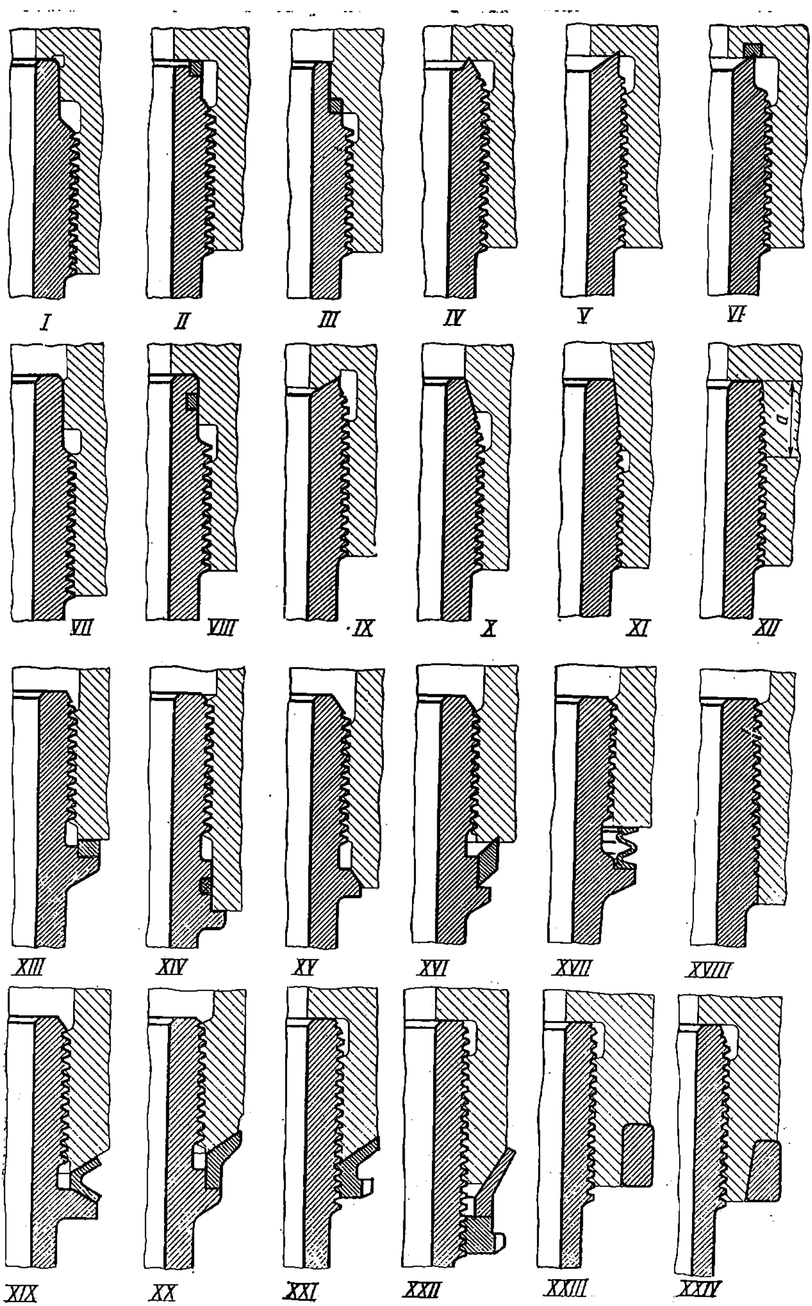


Fig. 327. Sealing tight-screwed assemblies

rately machined cylindrical portions (Fig. 327, *VII*, *VIII*); mating parts have a conical fit (Fig. 327, *IX*-*XI*). The design in Fig. 327, *XII* features the thread on the inner part cut away to a taper on portion *a*; the mating portion on the outer part is plain. When screwed in, the inner part cuts thread in the outer part.

The described joints will prove more reliable if the packing elements are disposed on the outside of the joint rather than on the inside, as shown in Fig. 327, *I*-*XII*, where the packing elements are subjected to high pressure. The outer arrangement is beneficial in that the pressure of the fluid being packed acts upon the packing elements only if the fluid breaks through the screw thread, where, in addition, the pressure drops considerably. Such joints are sealed with gaskets (Fig. 327, *XIII*, *XIV*), conical elements (Fig. 327, *XV*), circular edges (Fig. 327, *XVI*), bellows-type rings (Fig. 327, *XVII*), and incomplete threads (Fig. 327, *XVIII*).

In the designs according to Fig. 327, *XIX*-*XXII*, sealing is achieved by squeezing the extreme threads of the outer part with conical rings and nuts. The squeezing can also be effected with a ferrule press-fitted onto the outer part (Fig. 327, *XXIII*, *XXIV*) or with a clamping collar.

3.5. Sealing Various Joints Against Liquids

Sealing Cylindrical Joints. Fig. 328 shows instances of sealing cylindrical joints exposed to pressure of a liquid (e.g. liquid-cooled liners in piston-type internal combustion engines).

The simplest type of packing is a rubber O-ring placed in a packing groove cut in the liner (Fig. 328, *I*). In the free condition, the ring projects over the liner surface; as the liner is inserted into the cooling jacket, the ring becomes compressed and seals the interface between the liner and the jacket. For improved reliability, several packing rings may be arranged in tandem (Fig. 328, *II*).

A better variety of such a seal is illustrated in Fig. 328, *III*. Here, the packing grooves are made inclined in the direction opposite to the action of the liquid pressure. Under the pressure, the rings are forced into the narrow portion of the grooves and pressed against the surfaces being packed with a force proportional to the pressure. A groove that communicates with the atmosphere through a drain passage is arranged between the packing rings. The liquid escaping through the first ring gets out through the drain passage; the second ring, which is here free from the liquid pressure, prevents further escape of the liquid.

The sealing reliability is improved if several packing rings are used on the pressure side of the seal (Fig. 328, *IV*). Other shapes of packing grooves and rings are shown in Fig. 328, *V*. It is imperative

that the groove should have in all the cases a larger cross-section area than the ring; otherwise, rubber, which cannot practically be contracted (although it is deformable elastically to different shapes), would exert substantial radial pressures and so distort the liner to a bellmouth in a longitudinal section.

To ensure adequate pressure of the packing rings against the jacket wall, waved two-coil annular springs back up the packing rings from inside (Fig. 328, *VI*).

In some constructions, the seal is tightened axially. The method can be effected in the simplest way when the cooling jacket is de-

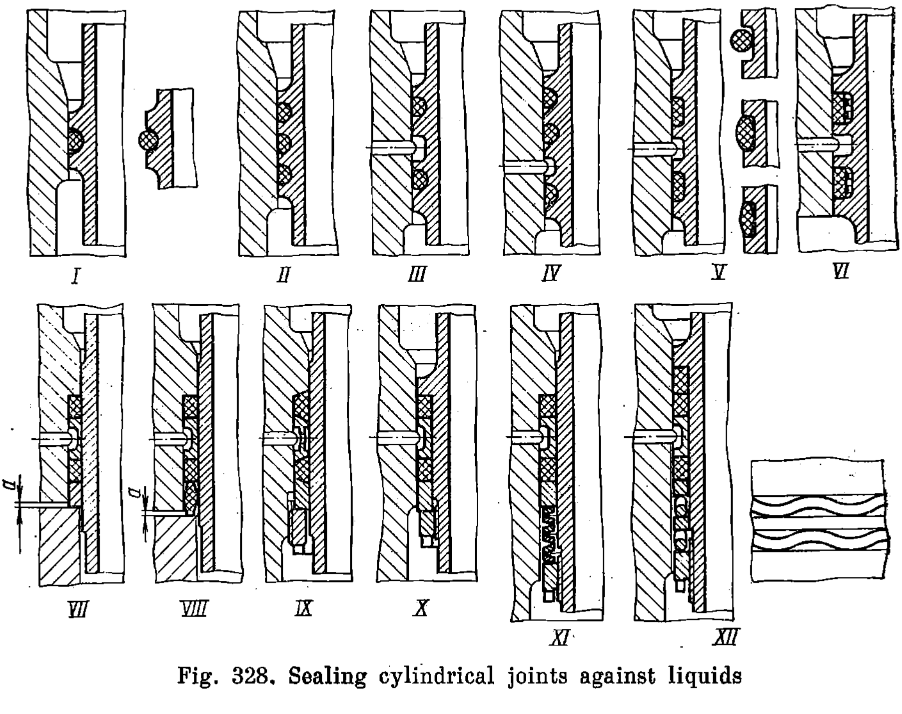


Fig. 328. Sealing cylindrical joints against liquids

tachable (Fig. 328, *VII*, *VIII*). The total length of packing elements is made here somewhat larger (by amount *a*) than the groove length; in tightening, clearance *a* will be made up for and axial tightness in the seal will arise.

With undetachable cooling jackets, a stuffing-box-type seal is used; the seal is tightened with a gland nut screwed into the jacket (Fig. 328, *IX*) or onto the liner (Fig. 328, *X*).

To provide against over-tightening the packing rings and maintain their constant compression in service, elastic elements, such

as Belleville spring washers (Fig. 328, *XI*) or waved spring washers (Fig. 328, *XII*), are introduced into seal units.

Sealing Flat Joints. A task to seal flat interfaces between cavities containing liquids and communicating with each other through complex-shaped or round openings is fairly common in engineering. Such joints are sealed with gaskets made from elastic sheet materials. Joints operating at high temperatures and pressures (e.g. the inter-

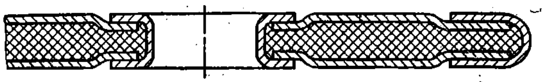


Fig. 329. Reinforced copper-asbestos gasket

face between the cylinder block and the cooling-jacket block in internal-combustion engines) are packed with reinforced asbestos gaskets.

The gaskets used come in two types: those with an inner frame and those encased in an outer shell. The gaskets of the first type are made from asbestos impregnated with a heat-resistant binding composition; the asbestos is pressed onto a copper or brass grid imparting to the packing element the required strength and stiffness.

The gaskets of the second type consist of an asbestos composition encased in a shell of thin-sheet copper or ductile iron (e.g. armco). The outer edges of the unit and all the openings are framed with the same material (Fig. 329).

3.6. Sealing Cylindrical Surfaces

Cylindrical joints assembled by interference fits need, as a rule, no sealing: the interference proper dependably seals the joint even at considerable pressure differentials. Subjected to sealing are transition- and clearance-fit assemblies exposed to fluid pressure or hydrostatic head. For instance, pistons and piston rods are joined together through face gaskets (Fig. 330, *I*, *II*) or elastic packing rings mounted in grooves cut in the mating cylindrical surfaces (Fig. 330, *III*). To prevent leakage of oil from the chamber being packed through the gap between a vertically disposed shaft and a spacer sleeve tightening the shaft bearing, the spacer sleeve is packed with an end-face gasket (Fig. 331, *I*, *II*) or with elastic rings located in appropriate grooves in the shaft portion mating with the sleeve hole (Fig. 331, *III*). Ring packings are also used to seal other parts of the joint where face gaskets are unsuitable.

Fig. 332 illustrates an application of rubber-ring packings for sealing an oil hole in an antifriction bearing-sleeve.

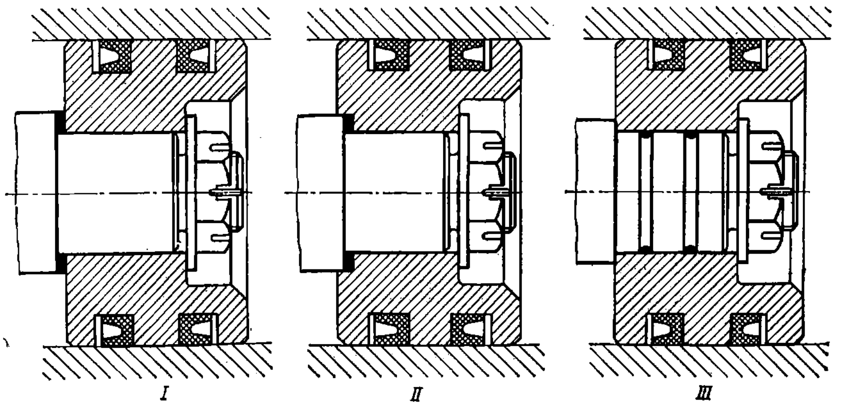


Fig. 330. Sealing cylindrical mating surfaces (piston-to-rod joint)

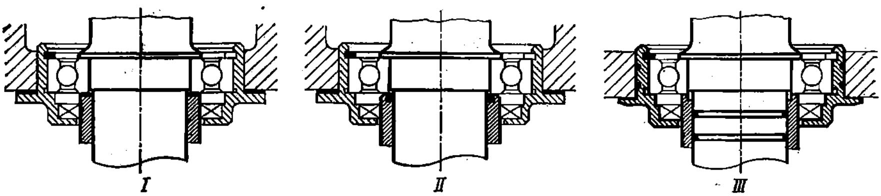


Fig. 331. Sealing cylindrical mating surfaces (vertical shaft)

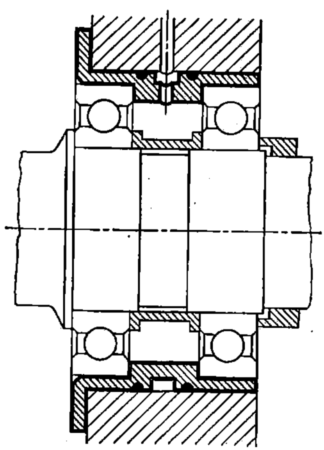


Fig. 332. Sealing oil hole with rubber rings in antifriction bearing unit

3.7. Sealing Quick-Removable Covers

The sealing of quick-removable covers, e.g., inspection windows or hinged slap doors, has some specific points. Commonly, the tightening pressure is here rather low and nonuniform (particularly, with hinged covers). Such covers are, as a rule, packed with gaskets made from thick, soft, and easy-to-compress materials (soft rubber, plastics, or cork). For convenient use, the gasket is secured to one of the joint faces by vulcanizing, by adhesive bonding, or by mechanical means.

Methods for sealing quick-removable covers are illustrated in Fig. 333. The designs in Fig. 333, *I*, *II* make use of a thick soft-rubber gasket vulcanized to the cover. In the designs shown in

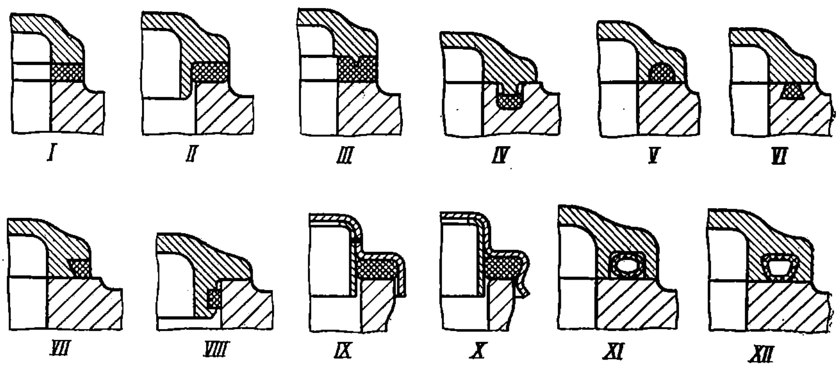


Fig. 333. Sealing quick-removable covers

Fig. 333, *III*, *IV*, the gasket is fixed to the housing. For higher reliability, covers are provided with a circular ridge, which is produced in round parts by turning on a lathe, and in profiled cast covers, by casting in permanent moulds. Rubber cord let into face or peripheral packing grooves can also be used (Fig. 333, *V-VIII*).

Fig. 333, *IX*, *X* shows stamped sheet-metal covers. Here, the gasket is secured in the annular pocket formed with the aid of an inner rim welded to the cover. Packings of improved elasticity, consisting of hollow tubular rubber rings filled with compressed air, are presented in Fig. 333, *XI*, *XII*.

3.8. Rubber as a Packing Material

Rubber used for sealing applications is almost always based on synthetic rubber materials, which, in comparison with natural rubber, exhibit high resistance to oil, benzine, and kerosene, and substantially outperform natural rubber in chemical, light, and temperature resistance.

Employed most extensively are chloroprene rubber, butadiene-styrene rubber, and butadiene-nitric rubber. Silicon rubber, which withstands temperatures up to 300 °C, is used in high-temperature applications.

Because of high elasticity, compliance, and ability to fill minute depressions and irregularities in the surfaces packed, rubber provides excellent packing properties.

Rubber is applied rather seldom as a sheet packing material because it is easily extruded under tightening pressure. It is advantageous to use sheet rubber (in the form of cord let into packing grooves, etc.) where the packing force is to be determined by its proper elasticity.

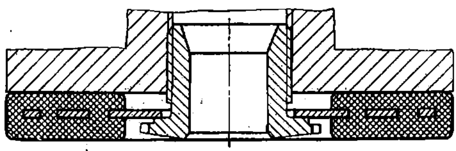


Fig. 334. Annular-disc packing unit

Sheet rubber is only suited to applications where the compression force is not high (e.g. for packing spring-loaded disc valves).

Securing rubber to a part presents some difficulties because of the tendency of sheet rubber to form creases. A method to obviate this disadvantage and, at the same time, to provide for reliable holding of a rubber sheet consists in application of reinforced rubber. This type of packing is obtained by pressing rubber on both sides of a metal sheet perforated in a staggered pattern. Rubber fills the perforations and so forms a secure bond with the metal sheet.

Improved bond of rubber with metal is obtained by applying a brass layer with a thickness of some hundredths of a millimetre to the metal surface. Metal sheet is coated with rubber cement, put into a rubber mixture, and subjected to simultaneous pressing and vulcanizing at a temperature of 140 to 150 °C and a pressure of 20 to 30 kgf/cm^2. In this way, stiff packings retaining all the advantageous properties of rubber are obtained.

An annular disc packing of this type is shown in Fig. 334. The packing is fastened to the part being sealed with a sleeve nut, whose shoulder bears against the inner portion of the metal disc extending from the rubber ring.

Fig. 335 shows disc-type valves with rubber packings. The rubber is secured to the disc metal surface by vulcanizing or adhesive bonding (Fig. 335, *I*). To bond rubber to metal, use is made of butadiene-styrene, neoprene and siloxane adhesives along with adhesives based on modified epoxy resins.

Fig. 335, *II*-*VI* displays mechanical fastening methods for rubber packings. The use of a metal washer (Fig. 335, *II*) has the disadvantage that the outer periphery of the rubber disc can come off the metal surface as the washer is pressed against the disc in tightening the cap screw. This disadvantage can be eliminated by placing the rubber disc in a circular recess with a beveled peripheral wall

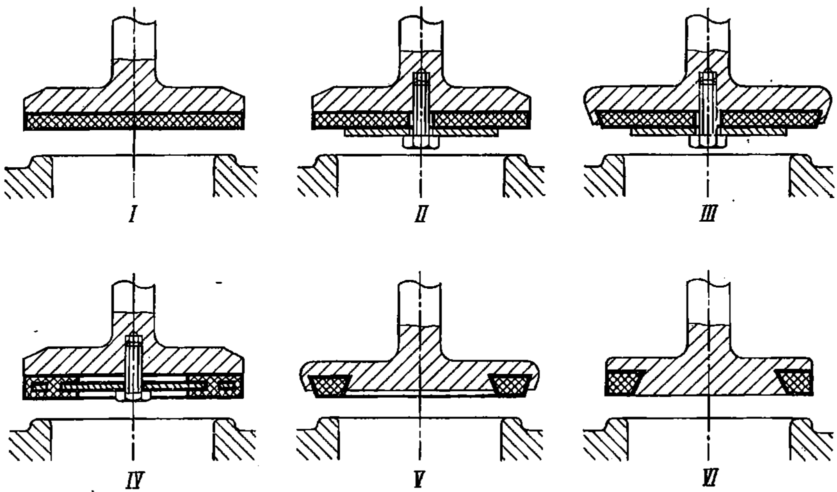


Fig. 335. Securing rubber to metal parts (lift-type valve with rubber packing)

(Fig. 335, *III*). Fig. 335, *IV* shows a packing similar to that shown in Fig. 334. In the design according to Fig. 335, *V* a rubber ring is fitted into an annular dovetail slot. The valve shown in Fig. 335, *VI* has a packing ring tightly set into an open peripheral groove.

Chapter 4

Springs

Springs are classified under the following groups: helical (coiled) springs, washer springs, annular springs, and leaf springs. Helical springs coiled from wire, normally of a round cross section (Fig. 336, *I*),

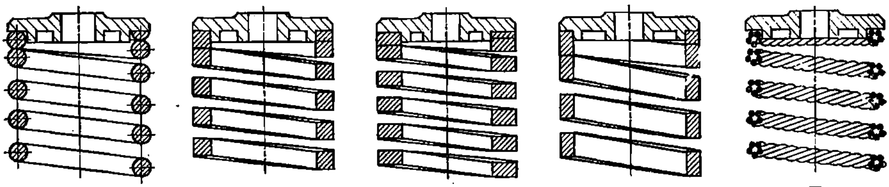


Fig. 336. Helical springs

and, for some applications, of a square or rectangular cross section (Fig. 336, *II* and *III*, *IV* respectively), are used most frequently. In some cases, springs are wound from multi-strand wire cables (Fig. 336, *V*).

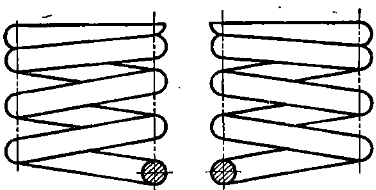


Fig. 337. Right-hand (*I*) and left-hand (*II*) coiled springs

With respect to their function, helical springs are divided into compression, extension, and torsion types.

Both right-hand and left-hand coil springs are used (Fig. 337). For helical compression and extension springs, the hand of coil is of no consequence. Washer springs and annular springs are used as compression springs only.

4.1. Materials and Manufacture

Springs are produced from carbon and alloy steels with a carbon content of 0.5 to 1.1 per cent. Carbon steels are used for springs up to 10 mm in diameter; alloy steels are used for springs working at high stresses and elevated temperatures and also for springs with large bar cross sections (from 20 to 30 mm in diameter) to ensure proper hardening throughout the whole section.

Addition of silicon (up to 2.0 per cent) improves the elastic properties of steel and its resistance to recurrent impact loads. Vanadium (0.1 to 0.2 per cent) and tungsten (up to 1.2 per cent) are introduced for higher mechanical properties and temperature resistance. Critical springs are manufactured from silicon-vanadium and chromium-silicon-vanadium steels having the highest mechanical properties.

Springs operating at elevated temperatures are produced from chromium-vanadium steels similar to Grade 50ХФА (temperature resistance up to 300 °C), silicon-tungsten steels, e.g., Grade 65С2ВА (up to 350 °C), and Grade 40Х13 steel (up to 450 °C).

Special steels with an increased content of Cr, V, Mo, and W are employed for springs working at temperatures over 500 °C.

Table 2 lists the main materials used for making springs and their mechanical properties after heat treatment. The modulus of elasticity in tension E of spring steels is equal to $(2.1\text{-}2.2) \times 10^4$ kgf/mm², and modulus of elasticity in shear G is equal to $(7.6\text{-}8.2) \times 10^3$ kgf/mm².

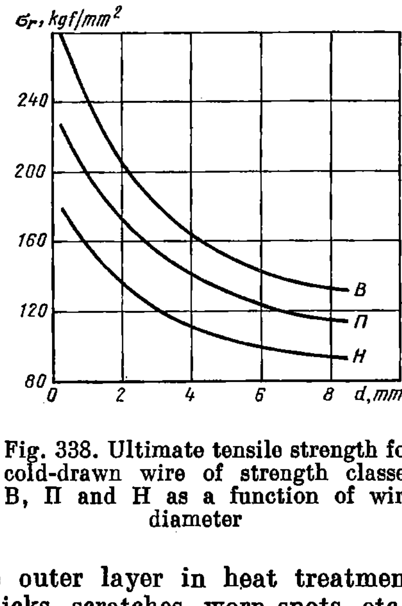


Fig. 338. Ultimate tensile strength for cold-drawn wire of strength classes В, П and Н as a function of wire diameter

The fatigue strength of spring steels depends little on their chemical composition and is far more dependent on the outer-layer condition. Decarburization of the outer layer in heat treatment and also local defects (corrosion, nicks, scratches, worn spots, etc.) sharply decrease the ultimate fatigue strength. The fatigue life of spring steels can be substantially extended through polishing and, especially, strain hardening of the surface layer by drawing, shot peening, etc.

For pulsating cyclic loads, the ultimate fatigue strength ranges from 40 to 60 kgf/mm².

The strength of spring steels greatly depends on the wire diameter, namely, it sharply rises with the decreasing diameter. Fig. 338 illustrates, by way of example, the strength characteristics of cold-rolled wire as a function of its diameter. The strength of small-diameter wire (d = 0.2 to 1.0 mm) is twice that of wire with a diameter of 8 mm. The wire diameter should be allowed for when selecting the permissible values of stress for spring calculation.

Table 2

Mechanical Properties of Spring Materials

Material	Grade	Ultimate strength in tension σ_t, kgf/mm²	Ultimate strength in torsion τ, kgf/mm²	Elongation δ, %
Carbon steels	65	100	80	9
	70	105	85	8
	75	100	90	7
	85	115	100	6
Piano wire	—	200-300	120-180	
Cold-rolled spring wire	Н	100-180	60-100	2-3
	П	120-220	70-130	
	В	140-280	80-160	
Manganese steels	65Г	70	40	8
	55ГС	65	35	10
Chromium-vanadium steel	50ХФА	130	110	
Corrosion-resistant steel	40Х13	110	80	12
Silicon steels	55С2			6
	60С2А	130	120	
	70С3А	180	160	5
Chromium-manganese steels	50ХГ		110	5
	50ХГА	130	120	6
Nickel-silicon steel	60С2Н2А	180	160	
Chromium-silicon-vanadium steel	60С2ХФА			5
Tungsten-silicon steel	65С2ВА	190	170	

Corrosion-resistant steel, grade 40Х13, or copper-base alloys are used for springs designed to work in highly humid and chemically aggressive environments. Table 3 presents the most widely applicable copper alloys and their mechanical properties.

The modulus of elasticity in tension E of copper-base alloys equals (1.2-1.3) $\times$ 10^4 kgf/mm², and the modulus of elasticity in shear G equals (4.5-5.0) $\times$ 10^3 kgf/mm².

Beryllium-bronze alloys are superior to other spring materials in corrosion resistance and fatigue strength. These properties along

Table 3

Mechanical Properties of Copper Alloys

Material	Grade	Ultimate strength in tension σ_t, kgf/mm²	Ultimate strength in torsion τ, kgf/mm²	Elongation δ, %
Gun metal Silicon-manganese bronze	БрОЦ 4-3 БрКМц 3-1	80-90	50-55	1-2
Beryllium bronze	БрБ2, БрБ 2.5	80-100	50-60	3-5

with high electrical conductivity make for wide application of beryllium bronze as a spring material in the electrical industry. Moreover, beryllium bronze shows highly stable elastic properties and almost total absence of hysteresis; for this reason, it is well suited to flexible components in precision instruments.

Springs made from copper-base alloys are paramagnetic; they are applicable where the influence of magnetic fields needs to be prevented.

Helical springs from a small-diameter wire or bar (up to 10 mm) with a ratio $D/d > 4$ (D is the mean coil diameter of spring and d is the diameter of spring wire) are produced by winding in cold condition. Springs with a ratio of $D/d < 4$ and also springs from large-diameter bars are wound hot.

There are two methods of cold winding:

(1) wire is wound either after heat treatment or after cold drawing and subjected to low tempering (200 to 300 °C) to relieve the stress that arises in winding;

(2) wire is wound in the annealed state and then hardened and tempered.

The first method is applied to springs manufactured from carbon steels, e.g., piano wire and cold-rolled wire of 0.2 to 8mm in diameter, and also from silicon-tungsten and chromium-vanadium steels.

Piano wire is produced from quality carbon steel (under 1.0 per cent carbon) and is subjected to isothermal hardening (heating to 870-950 °C) with subsequent conditioning in a bath of molten lead at 500 °C (patenting). After the heat-treating process, the wire is drawn to size; it acquires high strength properties ($\sigma_t = 300$ kgf/mm²) as a result of strain hardening.

Springs are produced similarly from cold-rolled wire, which is available in three strength classes: normal H, medium П, and

high B, with subdivision, depending on toughness, into groups *I* and *II* (for classes H and B), and *I-III* (for class П).

Alloy-steel springs (except those made from silicon-vanadium and chromium-vanadium steels), after being wound, undergo heat treatment: oil hardening at 800 to 850 °C and subsequent tempering at 400 to 500 °C.

To avoid decarburization of the surface layer, springs are covered before heating with a layer of charcoal powder or iron sawdust. The heat treating conditions are specified in detail for each grade of steel and should be strictly followed if the best strength characteristics are to be obtained.

Hot-coiled springs are always heat treated. Coiling is effected at 800 to 1,000 °C.

Springs are wound from grades БрОЦ 4-3 and БрКМц3-1 bronze and, after winding, are heated to 100-150°C for stress relieving. Beryllium-bronze springs are water-quenched from 800°C and tempered at 250 to 350°C.

Steel springs designed for vital applications under cyclic loads are shot-peened after heat treatment.

The final spring manufacturing operation is corrosion-protective coating. Steel springs are normally plated with zinc, cadmium, etc.; they can also be phosphatized.

4.2. Presetting

Helical compression springs whose coils are subject to torsion are strengthened by presetting. This strengthening method consists in producing in coil outer fibers, which are stressed most, an initial stress opposite in sign to the working stress.

Presetting involves compression of the spring by a load exceeding the working load; the presetting load is so selected that the shearing stress in coil outer fibers exceeds the elastic limit, and the material at these portions becomes deformed permanently (Fig. 339, *I*). The spring is held under load for 36 to 48 h, after which the load is removed.

The resilience of the coil inner fibers, which have not acquired residual deformations, gives rise to a shearing stress in the deformed layers, opposite in sign to the working stress (Fig. 339, *II*). An insignificant reactive stress of the same sign as the working stress arises in the core of the section. When the working load is applied to the spring (Fig. 339, *III*), the initial shear stress, added to the working stress, will result in a substantially lower magnitude of stress in the outer fibers than that in a spring not subjected to presetting (Fig. 339, *IV*). The reactive stress in the core, adding to the working stress, gives a total stress magnitude somewhat exceeding that in an unpreset spring.

As a result, the coil cross section turns out to be loaded more uniformly, and the material is utilized more adequately; the stress peak on the coil circumference is decreased. In short, the spring becomes stronger. With equal loads, the maximum stress in a preset spring is lower than in an unpreset spring, and with the maximum working stresses being equal, the former can sustain a higher load than the latter.

Owing to the residual deformations in the outer fibers, a spring acquires some permanent set; the coil pitch and the overall spring

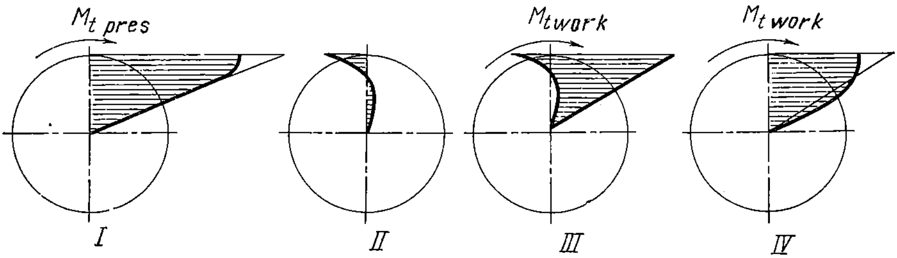


Fig. 339. Shear-stress distribution diagram for coil cross section of preset spring

length are reduced compared with those produced in winding. Therefore, when designing preset springs, the original coil pitch and spring length should be specified with regard to this reduction. No heat treatment is allowed after presetting since heating reduces or totally nullifies the obtained useful residual stresses.

The magnitude of the presetting load, amount of permanent set, and the extent to which the spring must be strengthened are determined by special design calculations or found by testing trial specimens.

The presetting is only applicable to springs subjected to static loading or periodical dynamic loading with a limited number of cycles. Springs operating at high-frequency cyclic loads over a long period of time are not subjected to presetting because resonant oscillations, which arise under these loading conditions, upset the stress pattern in the coils and cause increased stresses in preset springs.

4.3. Allowable Working Stresses

The allowable working stresses (design stresses) are specified with regard to the following factors:

—torsional or flexural strength of material (for compression/extension springs and for torsion springs, respectively);

—scale factor (wire diameter);

—wire-surface condition (surface roughness, work hardness, decarburization, etc.);

—spring service conditions (working temperature, corrosive environments, wear of coils, etc.);

—functional importance of the spring (consequences of breakage or lost compliance, hazard of accident from breakage, etc.);

—conditions of load application (amount of load eccentricity)

—type of loading.

Distinction is made between three types of loading.

Type I. Static Loading. Springs are subjected to a constant load or periodic loads with gradually changing magnitudes.

Examples are springs in locking and balancing mechanisms, braking, winding and accumulating springs, reduction- and safety-valve springs.

Type II. Dynamic Loading of Limited Duration. Springs are subjected to periodic impact loads or cyclic pulsating loads during a service period of not over 100,000 cycles.

Examples are springs in trigger assemblies of firearms, in shock absorbers of periodic action, in automatic machines, and low-speed cam mechanisms.

Type III. Dynamic Loading of Indefinite Duration. Springs operate under high cyclic loads during unlimited time.

Examples are springs in multi-revolution cam mechanisms, particularly, valve springs in internal-combustion engines, continuous-action shock absorbers in vehicles, springs in shakers and rotary forging machines.

With the type I loading, springs should be designed for the torsional or flexural elastic limit (extension/compression springs and torsion springs, respectively); here, a safety factor no less than 2.0 should be assumed. The type II loading also requires calculation for the elastic limit, but the safety factor should be increased from 1.3 to 1.5 times. Springs subjected to type III loading should be calculated for the ultimate fatigue strength, with a safety factor of 1.3 to 1.5. For critical applications, where spring breakage may lead to grave accidents, the safety factor is assumed to be from 2.0 to 2.5, i.e. such springs should have higher margin of safety.

The foregoing recommendations are largely theoretical. From them it follows, for instance, that non-critical statically loaded springs can operate at higher working stresses than the springs loaded otherwise. In practice, however, the lowest stresses at which the spring retains acceptable (from the design standpoint) dimensions are specified for such springs. In contrast, high design stresses are frequent in critical cyclically-loaded springs, e.g., in valve springs, which, in order to endure such stresses, have to be made from high-quality materials. This approach to the design of critical springs is dictated by the necessity to reduce as much as possible the overall dimensions and mass of the spring proper and the co-active parts that directly determine the magnitude of cyclic loads in the mechanism.

As a general rule, springs should be designed to work at the lowest stresses that meet the specified loading and compliance requirements within acceptable dimensions. Higher factors of safety provide against breakages due to unforeseen causes, such as unspecified changes in the process of spring manufacture and heat treatment, local defects in spring material that cannot sometimes be detected by a most careful inspection, etc.

Proper caution should be exercised when specifying increased stress values. Higher stresses not only involve some risk, but also lead to increased manufacturing costs because of the necessity to use expensive alloy steels, to follow strictly the heat-treating conditions, and to inspect, test, and accept springs very carefully. In this respect, checking against the characteristics of the designs proven during long service may be helpful.

Generally, the allowable working stresses in springs vary within the range of from 4,000 to 7,000 kgf/cm^2 (most frequently, within 4,000 to 5,000 kgf/cm^2). These values are fairly stable for general-purpose springs made from carbon or low-grade alloy steels (manganese and silicon steels) operating under static loads (type I loading) or under dynamic loads with a limited number of cycles (type II loading). Stresses in springs produced from quality steels (tungsten-silicon, chromium-silicon-vanadium, and chromium-vanadium steels), operating under durable dynamic loads (type III loading), are of the same order.

The higher stress limits are only resorted to when designing springs for use in confined spaces dictating reduced spring dimensions, for high-frequency cyclic loading of definite duration (as with the type II loading), which requires a reduced mass of the spring, etc. In some firearm-spring applications, for instance, the allowable stress limit is brought up to as high as 10,000 kgf/cm^2.

4.4. Design of Helical Compression Springs

4.4.1. End Coils

The proper function of compression springs is dependent on the form of their end coils, which should meet the following requirements:

—contact interface between the end coils and supporting parts must be flat and square with the spring axis to prevent the application of load at one point only;

—wherever possible, the contact area should be a complete coil to avoid the application of load out of centre;

—design of the end coils must provide for the correct centring of the spring in the supporting parts.

Figure 340, *I* shows a defective spring end obtained by merely clipping the wire. Such an end form results in a point contact and an out-of-central application of load. The spring is subject to bending; it will buckle or misalign depending on the mutual angular position of both clipped ends. The centring of the spring is difficult to achieve.

The design in Fig. 340, *II*, where a projecting end of the spring is ground flat, is also incorrect. The contact area is restricted; its extension depends on the coil lead angle (with the spring shown in the

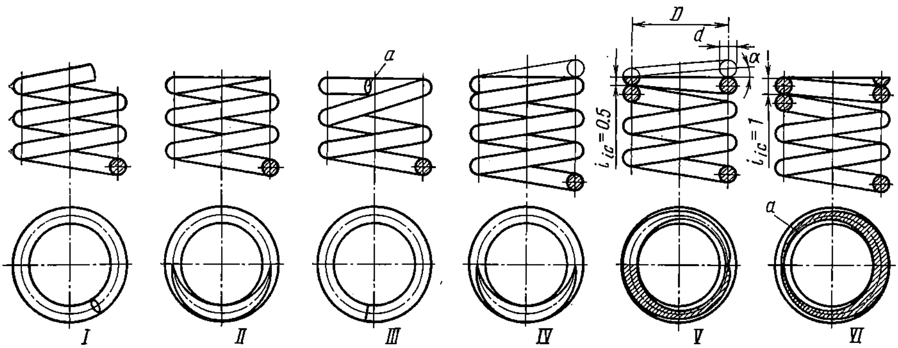


Fig. 340. Methods of shaping spring end coils

drawing, the contact-area angle is 180°). The load applied acts eccentrically; the thin projecting end is apt to break off.

In the spring presented in Fig. 340, *III*, the end coil is bent to the plane square with the spring axis. The design is seemingly close to meeting the requirement for a bearing plane of large angular extension. But an analysis of operation of the spring under load shows that the solution is inadequate. In fact, the lead angle of coils becomes reduced on applying the load; as a result, clipped end *a* rises above the bearing plane, the load concentrates at one point, and the coil portion disposed behind the clipped end will tend to break.

The spring shown in Fig. 340, *IV* has the closed ends squared by grinding. In this case, the extension of the bearing area is limited; it varies inversely with the coil lead angle.

The contact around the whole coil circumference can only be achieved if the closed coils are given a lead angle different from that of the rest of coils (found from the relationship $\tan \alpha = 2D/d$, where D is the mean coil diameter and d is the diameter of spring wire). To put it in a different way, the closed-coil cross-section axis on one side of a lengthwise spring section should be disposed against the contact point of adjacent closed coils on the opposite side of the section.

Figure 340, *V* shows a spring wherein a closed-end coil is ground away to half the diameter from the point of contact with the adjacent

working, or active, coil. The bearing area, shown in the plan view, comes out closed; it passes on one side along the closed coil (the hatched area on the plan view), and on the other side, along the active coil (the clear area on the plan view). Thus, the active coil here turns out to be substantially weakened.

To prevent the weakening of the nearest active coil, at least one whole coil should be closed (Fig. 340, *VI*). The bearing area will then be disposed entirely on the closed coil (the hatched area on the plan view); the adjacent coil will work over the whole cross section. The

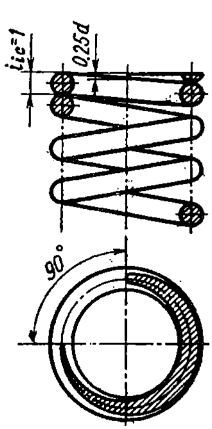


Fig. 341. Clipped thin end of the last coil

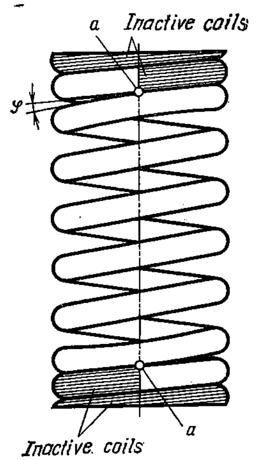


Fig. 342. Determining boundary between active and inactive coils

weakened closed coil is of no consequence as it bears on the adjacent active coil along the full length.

The thin end *a* of the closed coil (Fig. 340, *VI*, plan view) is undesirable since it can break off under load. In practice, the end is always removed around an arc of 90° (Fig. 341), and some eccentricity of the bearing area that arises from the removal is admitted. The minimum thickness of the closed end at the point of clipping becomes equal to $0.25d$. The clipped end is blunted around.

The closed end coils take practically no part in operation of a spring and have no influence on its compliance in contrast to active coils, which deflect under load.

As their number determines the compliance of a spring, the active coils should be clearly distinguished from the end, or inactive, coils. The characteristic feature of inactive coils is that they have no deflection with respect to the spring supporting members.

While the inactive coils of the free end of a spring deflect together with the spring cap, those of the stationary end are immovable.

Point a, i.e., the initial point of contact between coils in an unloaded spring (Fig. 342), is assumed to define the boundary between the active and inactive coils. Accordingly, the number of closed coils extending from point a to the point where the last coil comes to an end is taken as the number of inactive coils i_{ic}; the clipped end of the last coil is also included in the inactive coils.

The above distinction is rather conventional and somewhat indefinite because first, to spot point a where the coils come into contact is rather difficult owing to an insignificant divergence angle φ between the closed and the active coil and, second, point a shifts towards the active coils as the spring is compressed. So, with the de-

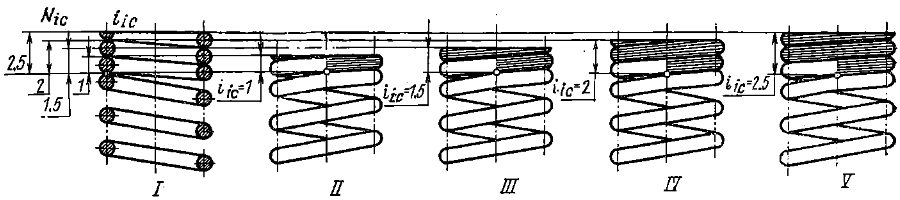


Fig. 343. Determining the number of inactive coils

flection of the spring the actual number of inactive coils increases and the number of active coils decreases.

A more correct approach would be determining the number of inactive coils under load; it is, however, difficult to do, and the above conventional distinction between active and inactive coils is what the designer is bound to accept in practice.

In the spring illustrated in Fig. 342, the number of inactive coils is equal, according to the above rule, to 1.5 for each spring end, or 3 for the whole spring. The number of active coils equals 7, and the total number of coils, 10.

As is indicated before, there should not be less than one inactive coil if the weakening of the adjacent active coil is to be avoided. In practice, 1.5 inactive coils on each side are adopted most commonly. With long springs and with springs subjected to cyclic loading, the number of inactive coils is increased to 2-2.5.

The ratio of active coils to the total number of inactive coils in a compression spring should not be less than 3.

Figure 343 illustrates a graphic method of determining the number of inactive coils. Fig. 344 is a front sectional view of springs having different numbers of inactive coils, the number of active coils being a whole (Fig. 344, *I*, *II*, *III*) and a multiple of 0.5 (Fig. 344, *IV-VI*).

It is worth noticing that for a whole number of active coils the representations of inactive-coil sections appear on one side of the spring axis, and for the number of active coils being a multiple of 0.5, identical inactive-coil sections are disposed crosswise.

In practice, to simplify the drawing work, the spring coil cross sections are represented schematically as in Fig. 345, *I*, which corresponds to one inactive coil, or, still simpler, as in Fig. 345, *II*,

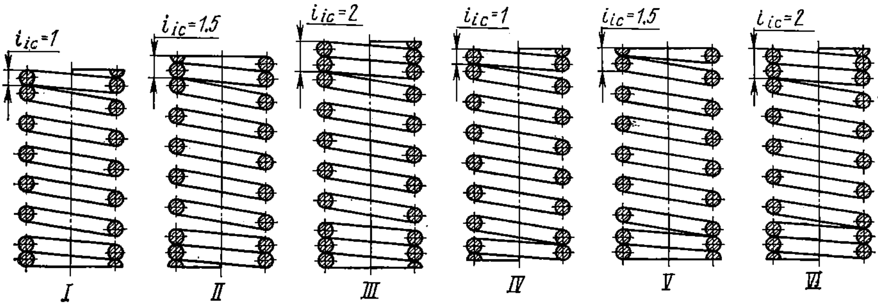


Fig. 344. Diagrammatic representation of springs with the whole number of active coils (*I-III*) and with the number of active coils being the multiple of 0.5 (*IV-VI*)

which corresponds to half the inactive coil. The actual number of inactive coils is indicated in an appropriate table on the drawing.

The spring ends are closed by different methods. With cold-wound springs, the end coils are set manually or on a mandrel having hel-

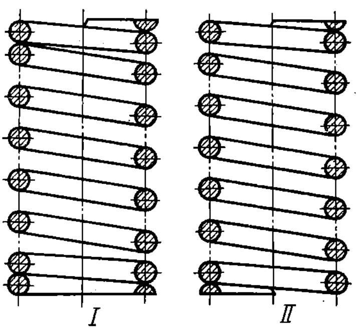


Fig. 345. Simplified diagrammatic representation of compression springs

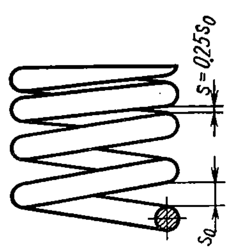


Fig. 346. Determining clearance between inactive coils

ical grooves with a pitch equal to that of the active coils; the mandrel is screwed into the spring so as to leave the ends to be closed free.

Tight closing of end coils is difficult to obtain by this method because after setting they spring back due to resilient forces. In practice, the closing is considered acceptable if clearance s between inactive coils does not exceed 0.25 of clearance s_0 between active coils (Fig. 346).

The changeable-pitch winding method, though more difficult to perform, gives better results. At the spring working portion, the pitch is rendered equal to that of active coils, and toward the ends, it is gradually reduced to the value corresponding to the spring wire diameter. This method makes it possible to obtain tightly closed inactive coils with some amount of interference between them.

In hot-wound springs, the end coils are also set in hot condition until they are completely closed.

4.4.2. Spring Centring Methods

Springs must be securely centred at both ends in assembly (Fig. 347). The centring is commonly done on the inner periphery of coils (Fig. 348, *I*). Centring on the outer periphery (Fig. 348, *II*) is only

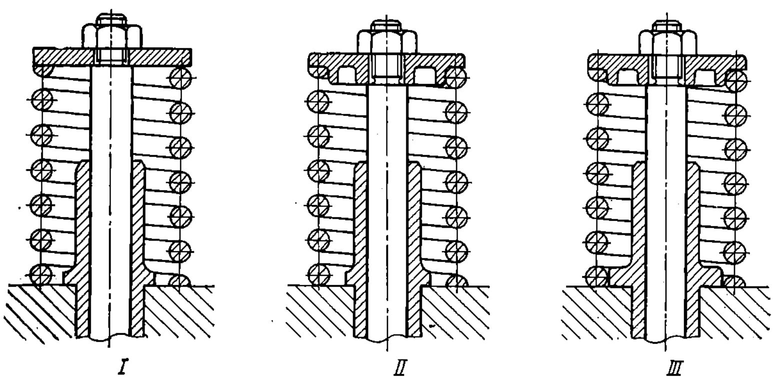


Fig. 347. Spring-centring methods
I and *II*—incorrect; *III*—correct

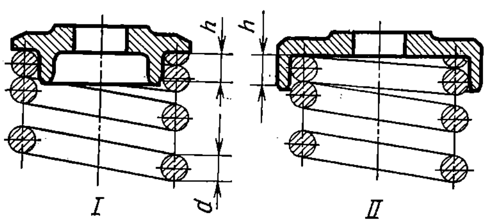


Fig. 348. Centring on inner and outer coil periphery

used where the spring is placed in some enclosing parts, such as housings or sleeves. To secure centring around the whole circumference of the spring, height h of the centring collar should at least be equal to d.

The inaccuracies of spring coiling dictate the use of a large-clearance centring fit: the clearance between the centring collars and spring coils is specified at 0.02 to 0.025 of the centring diameter. To prevent excessive friction between centring collars and spring coils, height h of a collar should be kept within $1.5d$. The clearance between coils and other centring-member portions contiguous to the centring collars should not be less than 0.3 to 0.5 mm (Fig. 349, *II*). This requirement is particularly important for springs housed in sleeve-type parts (Fig. 350) because the spring diameter increases as the spring is compressed.

To ease their insertion into springs with buckling or laterally oscillating coils, centring parts should be provided with entering chamfers.

The correct squaring of spring end coils includes chamfering the inner or outer coil periphery when the spring is centred on the inner or outer diameters, respectively (Fig. 351, *I*, *II*).

Sharp edges that arise in grinding the end coils are removed; otherwise, these edges will abut on the fillet joining the centring collar of a spring cap to its face (Fig. 352, *II*). In order that the spring cap might bear on the whole end-coil supporting area, the chamfer height must exceed the fillet radius; in this way, clearance a (Fig. 352, *I*) upsetting the correct bearing condition will not arise. In turn, the fillet radius must be smaller than the spring wire radius to ensure the correct bearing of the cap shoulder on the portion of the end coil having the full round cross section.

The chamfering is done with a conical abrasive tool. To provide against chamfer misalignment, the tool is centred on the inner or outer coil surface (for internally or externally centred springs, respectively).

By the use of specially-designed spring caps (Fig. 353, *I*), a compression spring with plain (not squared and closed) ends can be loaded centrally. The first, plain end a of the spring is screwed into a helical groove cut in the cap periphery, whose pitch equals that of the spring coils. Thus, the cap bears on the first coil along a whole turn of the helix (360° of arc), and, as the spring is compressed, the first coil moves together with the cap, i.e., works as a squared end coil. The next coil b is an active coil; it freely deflects during the spring compression, and its pitch and lead angle change accordingly. The spring cap shown in Fig. 353, *II* ensures still more uniform coil loading. The cap has a helical groove that accommodates two spring coils. The lead angle of the groove portion holding the first coil equals that of the latter; the lead angle of the groove portion that holds the second coil exceeds the coil lead angle so that a clearance s is formed between them, the clearance being equal to the whole deflection of the second coil under load. Thus, at the beginning of compression, the load is taken by the first coil; as the deflection is in progress, the second coil

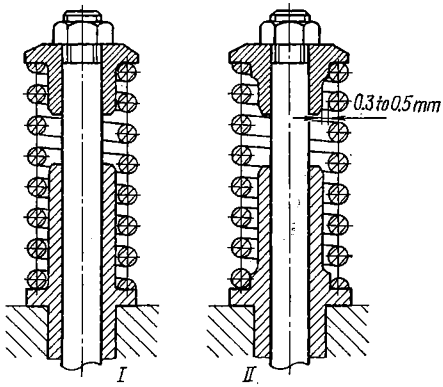


Fig. 349. Centring parts designed incorrectly (*I*) and correctly (*II*, *III*)

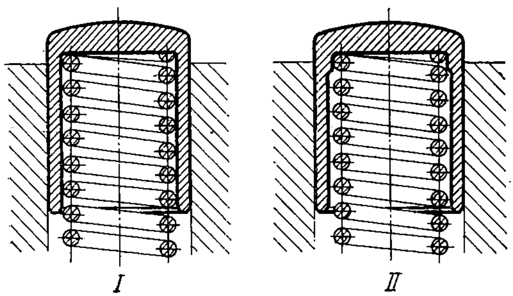


Fig. 350. Springs centred in sleeve-type parts
I—incorrectly; *II*—correctly

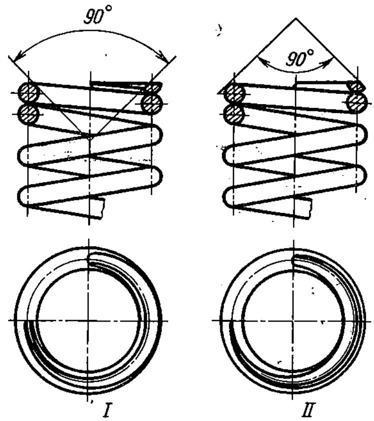


Fig. 351. Chamfered end coils

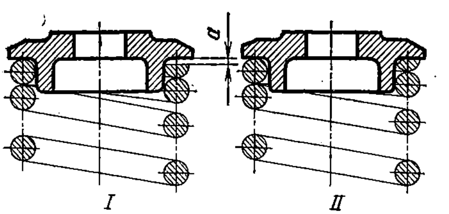


Fig. 352. End coil designed for correct contact with spring cap

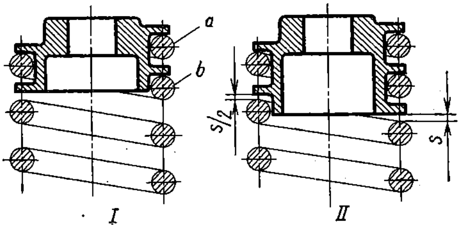


Fig. 353. Caps to accommodate plain-end springs

becomes loaded gradually, and at the conclusion of compression, the load is supported by the two coils.

Free and Working Lengths of a Spring. The length of a spring in its free, or unloaded condition (referred to as the free length) is given by

$$L = (i - 2)\, t + d\, (i_{ic} + 1) \tag{10}$$

where i, i_{ic} = number of active and inactive coils, respectively; t = pitch of active coils; d = spring wire diameter.

The length of a spring compressed solid is given by

$$L' = ad\, (i + i_{ic} - 1) \tag{11}$$

where a = factor allowing for incomplete closing of coils due to manufacturing defects, distorsion of coil helix, etc. (practically, $a = 1.1$ to 1.15).

During deflection, the spring diameter increases owing to the diminishing coil lead angle. The increase in the diameter is determined on the assumption that the wire length remains the same before and after deflection:

$$L_{wr} = \frac{\pi D i}{\cos\alpha} = \frac{\pi D' i}{\cos\alpha'}$$

whence

$$D' = D = \frac{\cos\alpha'}{\cos\alpha}$$

where D and D' = respective mean coil diameters of spring before and after deflection; α and α' = respective coil lead angles before and after deflection, found from the relationships:

$$\tan\alpha = \frac{t}{\pi D}; \quad \tan\alpha' \approx \frac{t'}{\pi D}$$

Here, t and t' = coil pitches before and after deflection ($t' = t - \lambda/i$, where λ = spring deflection).

The spring diameter can increase practically by several tenths of a millimetre, which should be taken into account when specifying the clearance value between the spring and the wall of a sleeve or any other part that will house the spring.

The decrease in the coil lead angle can also result in turning of the spring ends to a certain extent relative to each other, with the spring diameter remaining unchanged.

The length of each coil before deflection

$$l = \frac{\pi D}{\cos\alpha}$$

after deflection

$$l' = \frac{\pi D}{\cos\alpha'}$$

The difference in lengths

$$l-l'=\pi D\left(\frac{1}{\cos\alpha}-\frac{1}{\cos\alpha'}\right)$$

The difference for the whole spring (total difference)

$$\Delta=i(l-l')=i\pi D\left(\frac{1}{\cos\alpha}-\frac{1}{\cos\alpha'}\right) \tag{12}$$

Let us express this difference in terms of angle φ through which one spring end turns with respect to the other end (in radians):

$$\Delta=\frac{\varphi D}{2} \tag{13}$$

By equalizing expressions (12) and (13), we shall have

$$\varphi=2i\pi\left(\frac{1}{\cos\alpha}-\frac{1}{\cos\alpha'}\right)$$

$$\varphi=360^\circ i\left(\frac{1}{\cos\alpha}-\frac{1}{\cos\alpha'}\right) \tag{14}$$

Let us take, for example, a spring with $D = 50$ mm and $i = 10$; the coil pitch in the free condition $t = 10$ mm, and after deflection, $t' = 5$ mm.

Then,

$$\tan\alpha=\frac{t}{\pi D}=\frac{10}{\pi 50}=0.064$$

$$\tan\alpha'=\frac{t'}{\pi D}=\frac{5}{\pi 50}=0.032$$

i.e., $\alpha = 3°40'$; $\alpha' = 1°50'$.

From equation (14) we shall find the angle of mutual turning of the spring ends:

$$\varphi=360^\circ i\left(\frac{1}{\cos\alpha}-\frac{1}{\cos\alpha'}\right)=360^\circ\cdot 10\left(\frac{1}{0.998}-\frac{1}{0.9998}\right)\approx 7^\circ$$

4.4.3. End-Jointed Springs

For proper function of a compression spring, misalignment and lateral shifts of spring ends must be excluded. An example may be provided by the use of a compression spring as a return element for a pivoting lever (Fig. 354).

The construction shown in Fig. 354, *I* is faulty. Because of misalignment and mutual displacement of the spring-supporting surfaces during pivoting movement of the lever, the spring is subjected to lateral forces that cause additional bending stresses; the slanting spring axis (Fig. 354, *II*) reduces the bend of the spring to some extent.

In the design shown in Fig. 354, *III*, the spring-supporting surfaces are positively guided with respect to each other; the lateral forces are taken by the guide member, and the spring is subject to compression only. A spring can be prevented from buckling by mounting its support caps in ball-and-socket joints (Fig. 354, *IV*). The mutually guided spring caps (Fig. 354, *V*) increase the resistance of

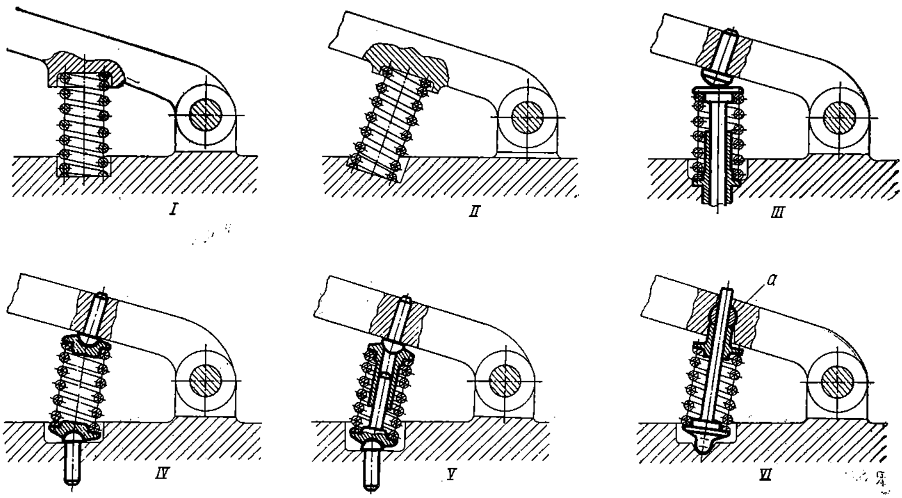


Fig. 354. Methods for coupling compression spring with pivoting lever

the spring to buckling. In the design presented in Fig. 354, *VI*, the spring is also mounted in guided caps; the upper cap abuts against a turning pin *a* placed in the lever body, and the lower cap bears on the base through a spherical joint.

In the mechanisms illustrated in Fig. 354, *IV-VI*, relative displacements of supporting points during pivoting of the lever are compensated for by changed angles of inclination of the spring axis. The arrangements of the above types are more suitable for extension springs as these are capable of self-adjustment with respect to their suspension points.

Figure 355, *I-VIII* shows pivots used where a spring can buckle in one plane only, and Fig. 355, *IX-XVI* shows ball-and-socket joints providing against buckling in all lateral directions.

In the design illustrated in Fig. 355, *I*, the spring cap has a cylindrical head that rests against the concave cylindrical surface of a supporting member. Here, the spring needs to be fixed laterally, i.e., along the cylinder axis, which can be achieved by placing the spring between two guide plates square with the axis of the supporting cylindrical surfaces.

In the designs shown in Fig. 355, *II-IV*, the cylindrical supporting elements are interlocked, which ensures lateral fixing of the spring in all directions (provided that the supporting surfaces are

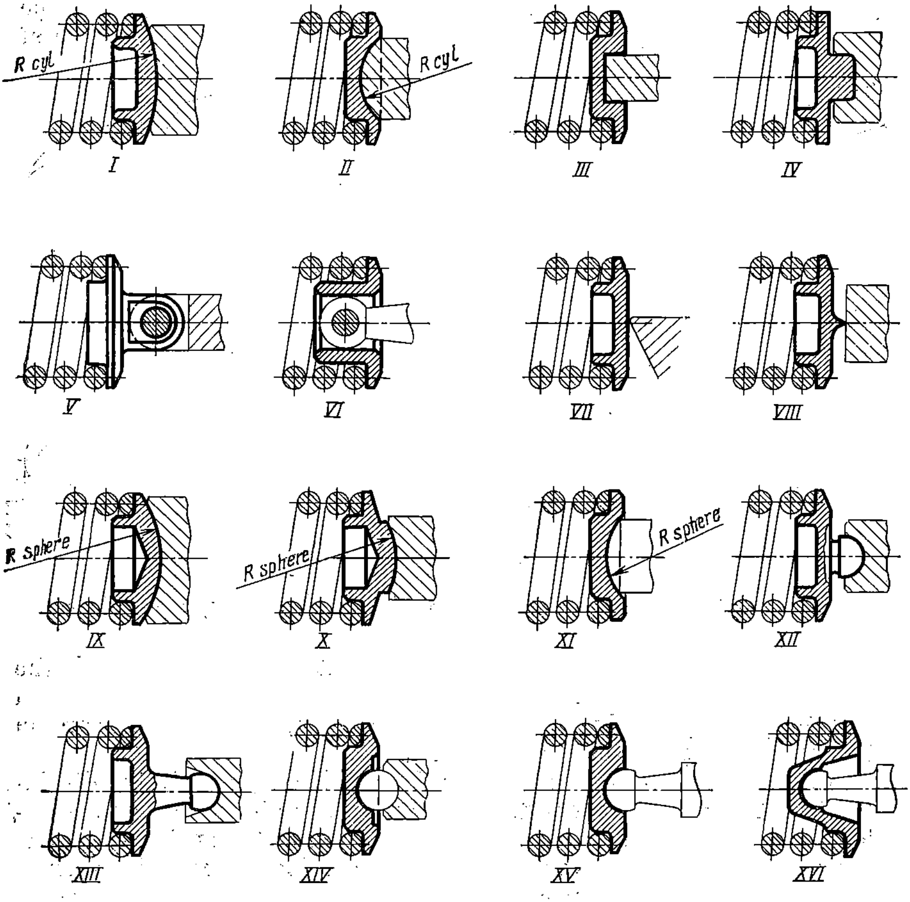


Fig. 355. Pivots and spherical joints for compression-spring ends

constantly spring-pressed against each other). Spring caps can also be pivoted on cylindrical pins (Fig. 355, *V*, *VI*). The design in Fig. 335, *VI* is suitable for longer springs as it provides greater steadiness of a spring in operation.

Fig. 355, *VII*, *VIII* illustrates location of spring caps on knife-edge supports. Here, the caps need to be fixed in all lateral directions.

Spherical joints are presented in Fig. 355, *IX-XVI*. A ball-and-socket joint placed inside a spring (Fig. 355, *XVI*) offers the highest axial stability. This design, however, is not recommended for application where the supporting points shift transversely relative to each other as the spring operates: the spring-axis angle of inclination will here be larger than in constructions where the supporting point is spaced apart from the spring (as shown, for instance, in Fig. 335, *XIII*).

Commonly mounted in spherical joints are precision springs, e.g. in measuring instruments, where some adverse factors, such as lateral forces, can cause distortion of spring characteristics.

4.4.4. Design Calculation of Springs

A typified calculation of a compression or extension spring is based on the assumption that the load is directed along the spring axis (Fig. 356). In this case, the forces acting on a coil at any section can be presented as a transverse force P, which bends the coil, and a torsional moment $M_t = PD/2$, which twists the coil. The bending by force P plays a secondary part; the effect of the torsional moment is prevalent, and, therefore, the spring is calculated for torsion.

Shear stress is at a maximum along the coil-section circumference and is determined by the known formula for a round-section beam subjected to torsion

$$\tau = \frac{M_t}{W_t} \tag{15}$$

where $W_t = \frac{\pi d^3}{16} \approx 0.2d^3$ = polar section modulus of spring wire (d = wire diameter).

The effect of the coil-axis curvature is taken into account by introduction of a stress-correction factor k, which is dependent on ratio $c = D/d$, called the *spring index*. With factor k being inserted into formula (15), the latter takes on the following form:

$$\tau = k\frac{M_t}{W_t} = k\frac{8PD}{\pi d^3} \tag{16}$$

The values of factor k as a function of spring index c are given in Fig. 357 for different coil lead angles α ($\tan \alpha = t/\pi D$, where t is coil pitch).

For common values of α taken at 6 to 12°, factor k is expressed fairly accurately as

$$k = \frac{4c+2}{4c-3} \tag{17}$$

The spring index c normally varies from 8 to 12, and factor k, corresponding to it, is equal to 1.1-1.2. These values can be taken for trial calculations.

Springs with an index $c < 4$ are not advisable to employ. Coiling such springs is difficult; cracks and fractures can occur in coil outer fibers, and working stresses in such springs are rather high.

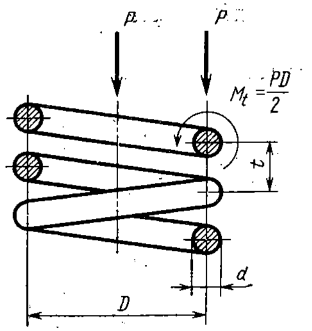


Fig. 356. Forces acting on axially-loaded compression spring

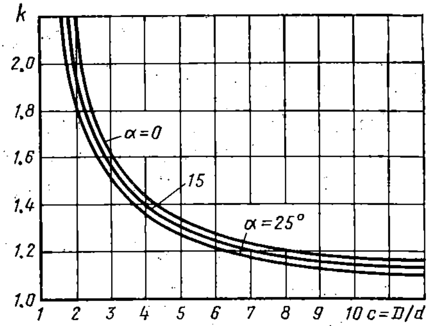


Fig. 357. Relationship between stress-correction factor and spring index for round-wire springs

From formula (16) it follows that the force exerted by the spring is

$$P = \frac{\pi d^3}{8kD}\tau = 0.392\frac{d^3}{kD}\tau \tag{18}$$

The spring deflection due to force P is given by

$$\lambda = \frac{8PD^3 i}{Gd^4} = \frac{8Pc^3 i}{Gd} \tag{19}$$

where i = number of active coils; c = spring index; G = modulus of elasticity in shear (for spring steels, $G \approx 8 \times 10^5$ kgf/mm^2, approximately).

Substituting the expression for P from formula (18) into formula (19), we obtain

$$\lambda = \frac{\pi D^2 i}{kGd}\tau = \frac{\pi cDi}{kG}\tau \tag{20}$$

The compliance of the spring is characterized by a parameter λ', which is deflection per coil under a load of 1 kgf.

From equation (19)

$$\lambda' = \frac{8D^3}{Gd^4} = \frac{8c^3}{Gd} \approx 10^{-5}\frac{c^3}{d}$$

The reciprocal of λ'

$$\delta = 10^5 \frac{d}{c^3}$$

is referred to as the *spring rate*.

The potential energy accumulated by the spring during its deflection is given by

$$U = \frac{P\lambda}{2}$$

The formulas (16) and (18) allow all spring parameters to be determined. The spring-calculation problem may be stated in two different ways. One approach is to calculate the stresses arising in coils and deflection due to force P by using known values of D, d, and i. More often, however, it is necessary to determine D and d from the given force P and spring deflection λ, provided that stress in the coils will not exceed the allowable value (τ_{al}).

In this case, the sequence of calculation is as follows:

1. A trial spring index value $c = 8\text{-}12$ is selected and factor k is determined from the diagram shown in Fig. 357 or from formula (17).

2. A trial mean coil diameter D of the spring is assumed for dimensional reasons.

3. The wire diameter d is determined by formula (18), assuming the value of τ_{al} to be within 4,000 to 6,000 kgf/cm^2:

$$d = \sqrt[3]{\frac{k8PD}{\pi\tau_{al}}} = 1.37 \sqrt[3]{\frac{kPD}{\tau_{al}}}$$

or

$$d = \sqrt{\frac{k8Pc}{\pi\tau_{al}}} = 1.6 \sqrt{\frac{kPc}{\tau_{al}}}$$

The found value of d is rounded off to the next greater standard value for the given grade of wire.

4. The spring index is computed and compared with the trial value. If necessary, the wire diameter is recalculated, using the value of c found in the preceding calculation.

5. The number of active coils required for obtaining the specified deflections λ is determined by formula (19):

$$i = \frac{\lambda G d^4}{8PD^3} = \frac{\lambda G d}{8c^3 P}$$

The obtained number of coils is rounded off to a whole number or a number that is the multiple of 0.5.

6. The length of the spring in the loaded condition (the working length) is found by the formula

$$L_{cp} = t_{cp}\,(i - 2) + d\,(i_{ic} + 1)$$

where i_{ic} = number of inactive coils, assumed to be 2 to 3; $t_{cp} = d + s$ = pitch per coil of the loaded spring (d is the wire diameter and s is a minimum clearance between active coils, specified at 0.3 to 1.0 d).

If the spring working length turns out to be unacceptable, the calculation should be repeated, assuming a larger spring diameter D.

At the final design stage, the free length of the spring is computed:

$$L = L_{cp} + \lambda$$

The active-coil pitch in the unloaded spring is determined by the following formula

$$t = \frac{L - d\,(i_{ic}+1)}{i-2}$$

The computed value is rounded off to the next pitch value that can be obtained by winding the spring on a lathe or an automatic winding machine. A diagram of the deflection-load relationship for the spring is then made; in this case, it takes the form of a right triangle (Fig. 358) wherein one leg represents spring deflection λ (in mm), and the other, on a suitable scale, force P (in kgf) corresponding to this deflection.

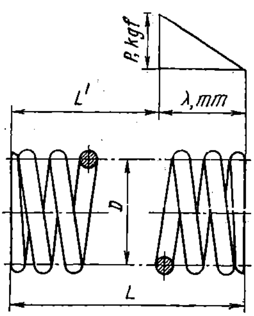


Fig. 358. Deflection-load relationship for compression spring

The solid length of the spring must be checked by using formula (11):

$$L' = (1.1 \text{ to } 1.15)\, d\,(i + i_{ic} - 1)$$

The developed length of the spring is then computed by the formula

$$L_d = \pi D \left(\frac{i}{\cos\alpha} + \frac{i_{ic}}{\cos\alpha_0}\right)$$

where i = number of active coils; i_{ic} = number of inactive coils; α = lead angle of active coils ($\tan\alpha = t/\pi D$); α_0 = lead angle of inactive coils. Here,

$$\left(\tan\alpha_0 = \frac{d}{\pi D} = \frac{1}{\pi c}\right)$$

Since $\cos\alpha_0$ is close to unity,

$$L_d \approx \pi D \left(\frac{i}{\cos\alpha} + i_{ic}\right)$$

or, almost equally accurate,

$$L_d = \pi D\,\frac{i + i_{ic}}{\cos\alpha}$$

To determine the length of the wire needed for coiling the spring, the value of L_d should be increased by 5 to 10% (allowance for end losses). Hence, for commonly applicable values of α = 6-12°, a sim-

plified formula may be used:

$$L_d \approx \pi D i_t$$

where i_t = total number of coils.

As a rule, compression springs operate under an initial load. The values of initial load P_{in}, maximum working load P_{max} and working

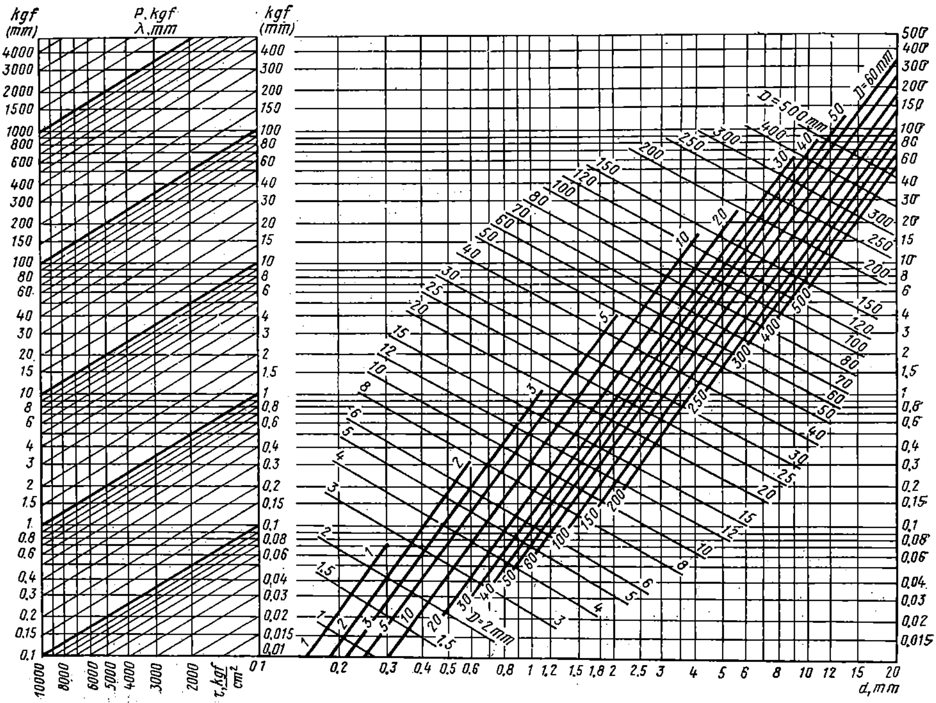


Fig. 359. Synoptic spring-calculation chart

deflection λ_w of a spring are usually established by specifications. In such cases, the spring calculation aims at selecting a spring with a deflection-load relationship that will allow the required values of P_{in} and P_{max} to be obtained for the given value of λ_w.

The order of the calculation is as follows. First, the spring parameter values necessary to secure the specified value of P_{max} are found; then, the total deflection λ_{max} of the spring is determined; and, finally, the value of P_{in} is computed from the equation

$$P_{in} = P_{max} \frac{\lambda_{max} - \lambda_w}{\lambda_{max}}$$

If the found value of P_{in} considerably differs from the specified value, the spring parameters should be changed until the desired deflection-load relationship is obtained.

A synoptic chart developed on the basis of equations (17) and (18), and (20) for the calculation of compression springs produced from wire of round cross section is presented in Fig. 359. Spring diameter values d are plotted on the abscissa. A set of heavy lines represents graphically the forces exerted by a spring at different values of its mean coil diameter D. The force values as dependent on torsional stress τ in kgf/cm^2 are given in the left-hand section of the chart.

A set of fine lines represents deflection per coil λ (in mm) as a function of torsional stress (in kgf/cm^2) at different values of mean coil diameter D. The deflections are determined by the same straight lines as the forces in the left-hand section of the chart.

The chart is developed for $G = 8 \times 10^5$ kgf/cm^2.

Let us consider several examples of use of the chart.

Example 1. Find stress in a spring with $D = 30$ mm and $d = 2.5$ mm loaded with force $P = 10$ kgf.

Solution. Draw a horizontal through the point of intersection of the ordinate $d = 2.5$ mm with the load curve for $D = 30$ mm (set of heavy lines) until it intersects the slanting line for $P = 10$ kgf at the left-hand section of the chart. The stress sought will be 5,600 kgf/cm^2.

Example 2. Determine stress in a spring with $D = 60$ mm and $d = 10$ mm deflected by 40 mm. The number of active coils is 8.

Solution. The deflection per coil is 5 mm. Draw a horizontal through a point of intersection of the ordinate $d = 10$ mm with the deflection curve for $D = 60$ mm (set of fine lines) until it intersects the slanting line for $\lambda = 5$ mm in the left-hand portion of the chart. A stress of 4,600 kgf/cm^2 is obtained.

Then draw a horizontal through the point of intersection between the ordinate $d = 10$ mm and the load curve for $D = 60$ mm (set of heavy lines) up to its intersection with the ordinate $\tau = 4{,}400$ kgf/cm^2 in the left-hand portion of the chart. The force exerted by the spring will be 220 kgf.

Example 3. Determine the deflection of a spring having $D = 30$ mm, $d = 2$ mm and the number of active coils, 8. The spring is subjected to a stress of 3,000 kgf/cm^2.

Solution. Draw a horizontal through the point where the ordinate $d = 2$ mm intersects with the deflection curve for $D = 30$ mm (set of fine lines) up to the intersection with the ordinate $\tau = 3{,}000$ kgf/cm^2 in the left-hand portion of the chart. The found deflection per coil is equal to 5 mm. The total spring deflection is 40 mm.

To determine the load on the spring, draw a horizontal through the point of intersection between the ordinate $d = 2$ mm and the load

curve for $D = 30$ mm (set of heavy lines) until it intersects the ordinate $\tau = 3{,}000$ kgf/cm^2; the load equals 2.8 kgf.

Example 4. Find the deflection of a spring with $D = 10$ mm and $d = 1$ mm loaded by force $P = 1$ kgf. The number of active coils is equal to 10.

Solution. Draw a horizontal through the point of intersection of the ordinate $d = 1$ mm with the load curve for $D = 10$ mm (set of heavy lines) until it intersects the slanting line for $P = 1$ kgf in the left-hand part of the chart; the stress is 3,000 kgf.

Draw a horizontal through the point of intersection of the ordinate $d = 1$ mm with the deflection curve for $D = 10$ mm (set of fine lines) up to its intersection with the ordinate $\tau = 3{,}000$ kgf/cm^2; the obtained deflection per coil is equal to 1 mm. Hence, the total deflection of the spring is 10 mm.

Example 5. Select a spring for working load $P = 40$ kgf and torsional stress $\tau = 4{,}000$ kgf/cm^2. The spring diameter $D = 40$ mm; the spring should be deflected by 20 mm.

Solution. Draw a horizontal through the point $P = 40$ kgf at stress $\tau = 4{,}000$ kgf/cm^2 until it intersects the load curve for $D = 40$ mm (set of heavy lines). The found wire diameter d is equal to 5 mm.

Draw a horizontal through the point of intersection of the ordinate $d = 5$ mm with the deflection curve for $D = 40$ mm (set of fine lines) up to its intersection with the ordinate $\tau = 4{,}000$ kgf/cm^2 in the left-hand portion of the chart; the found deflection per coil is 4 mm. Hence, the spring should have 5 active coils.

The free length of the spring is determined by formula (10), assuming the pitch of coil $t = 1.5\lambda + d = 11$ mm, and the number of inactive coils $i_{ic} = 3$; the sought length will be 53 mm.

Square or Rectangular Wire Springs. Stress in a spring with square cross-section coils is

$$\tau = k\,\frac{2.4PD}{a^3} \tag{21}$$

where D = mean coil diameter of spring; a = side of square; k = stress-correction factor equal to

$$k = \frac{4c+3}{4c-2}$$

where $c = D/a$ (Fig. 360).

The force exerted by the spring is

$$P = 0.416\,\frac{a^3}{kD}\,\tau \tag{22}$$

The deflection caused by load P (for the most common values of $k = 1.1$ to 1.2)

$$\lambda = 5.57\,\frac{PDi}{Ga^4} = 5.57\,\frac{Pc^3 i}{Ga} \tag{23}$$

or

$$\lambda = 2.3\frac{Di\tau}{Ga} = 2.3\frac{Di\tau c}{G}$$

where i = number of active coils; G = modulus of elasticity in shear.

Comparison of expressions (22) and (23) with expressions (18) and (19), respectively, shows that at equal magnitudes of stress, a square-

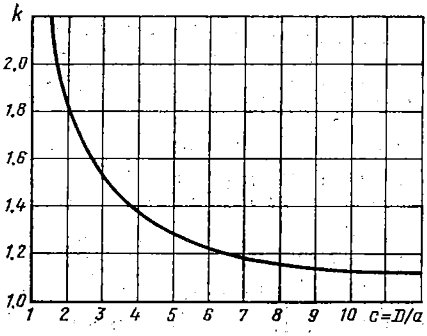


Fig. 360. Relationship between stress-correction factor and spring index for square-wire springs

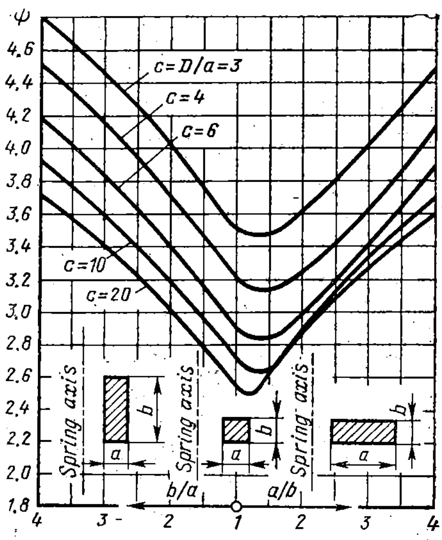


Fig. 361. Factor ψ as a function of b/a for different values of spring index of rectangular-wire springs

wire spring exerts a 3% greater force and has an about 30% smaller deflection than a round-wire spring with a diameter $d = a$.

Stresses in coils of a rectangular section with sides a and b (a is the side perpendicular to the spring axis) can be found from the relationship

$$\tau = \psi\frac{PD}{ab\sqrt{ab}} \tag{24}$$

The values of factor as a function of b/a for different values of $c = D/a$ are presented in Fig. 361.

Springs with a coil cross-section wherein the longer side extends along the axis ($b/a > 1$) exhibit higher rigidity and are used to support increased loads with small deflections.

Springs wherein the longer section side extends in a direction perpendicular to the axis ($a/b > 1$) have a slightly sloping load-deflection curve. It should be noted that the manufacture of such springs is difficult because of large flexural deformations arising in the course of winding. The cross sections with $a/b > 2$ are not recommended for use.

The spring index $c = \frac{D}{a}$ should not be less than 4 in all cases.

4.4.5. *Stability of Springs*

Long springs tend to buckle, i.e. lose longitudinal stability.

For axially-loaded springs with properly guided centring members ensuring parallelism of spring end coils during working movement,

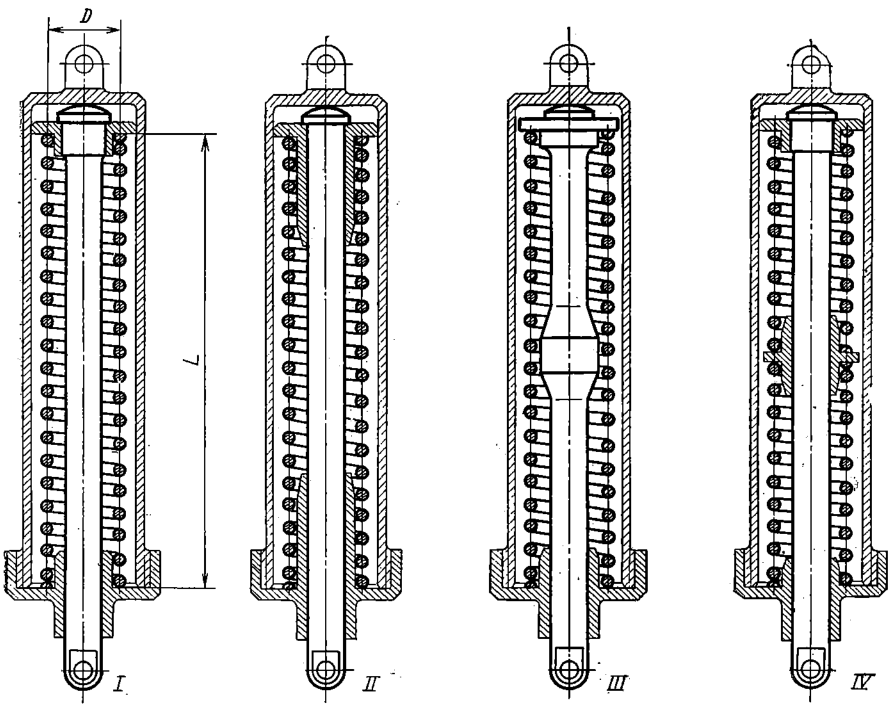


Fig. 362. Springs protected from buckling

the maximum ratio of free length L to mean coil diameter D (Fig. 362, I) at which the spring still retains its stability is approximately equal to 5; for springs with articulated centring members, $L/D = 2.5$.

Making allowance for the possibility of load applied out of centre, for distorsion of coils, etc., it is advisable to reduce the L/D ratio by 35 to 50% as compared with the above values.

For horizontally-disposed or inclined springs, where transverse loads can arise from the weight of coils, the L/D ratio should be reduced still further.

If the use of long springs cannot be avoided, special measures against the loss of stability should be taken. Longer centring collars for a reduction of the unguided length of the spring (Fig. 362, *II*) or an additional centring collar in the spring middle portion (Fig. 362, *III*) tend to increase the friction and, consequently, wear of coils. The best solution to the problem is to divide long springs into several short and steady ones supported by floating centring members appropriately guided along the spring axis (Fig. 362, *IV*). This approach, however, entails increase in the overall length of the unit because here additional end coils are needed to support the floating centring parts.

4.4.6. Resonant Oscillations

Springs functioning under cyclic loads are subject to resonant oscillations, which increase stress within coils and distort elastic characteristic of the system. Resonant oscillations are the most common

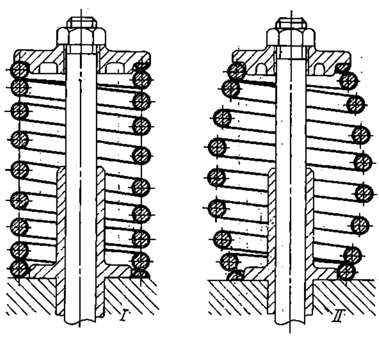


Fig. 363. Spring designs proof against resonant oscillations

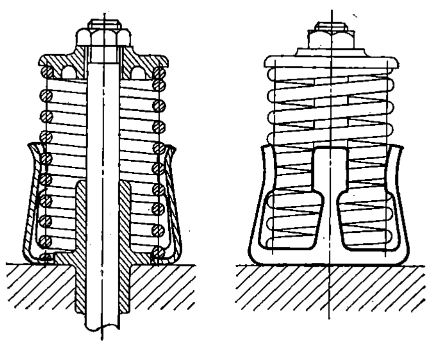


Fig. 364. Oscillation absorbers mounted on compression springs

cause of breakages in springs subjected to high-frequency cyclic loading.

Such springs must be so designed as to obtain the values of parameters (D, d, L_n, i) that will prevent the occurence of resonant oscillations in service. The methods of the design calculation are disclosed in special literature.

To avoid resonant oscillations, use is often made of springs with a variable coil pitch (Fig. 363, *I*) and of specially-shaped springs, e.g. barreled springs (Fig. 363, *II*).

An effective method of preventing resonant oscillations is the use of oscillation absorbers, e.g. in the form of thin leaf springs that touch the coils at the portions where the amplitude of oscillations is maximum (Fig. 364).

4.4.7. Multi-Spring Systems

Sets of concentrically disposed springs (Fig. 365, *II*, *III*) are used for double purpose: (1) to increase the system's compliance (for the

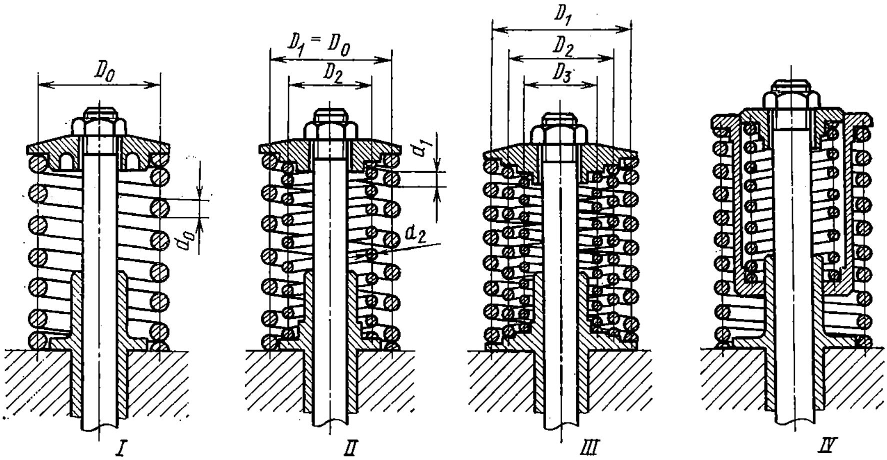


Fig. 365. Multi-spring systems

specified radial dimensions and load P); and (2) to increase the load-supporting capacity.

The maximum stresses and working deflections in a system of concentric springs should be identical. This condition results in a relationship

$$i_1 d_1 = i_2 d_2 = \ldots = i_n d_n \tag{25}$$

or

$$\frac{d_1}{d_2} = \frac{i_2}{i_1}, \quad \text{etc.} \tag{26}$$

where i and d = number of coils and wire diameter, respectively; n = number of springs in the set.

As can be readily shown, both these conditions are met if the indexes of the concentric springs are equal:

$$c = \frac{D_1}{d_1} = \frac{D_2}{d_2}, \quad \text{etc.}$$

Hence,

$$d_1 = \frac{D_1}{c}; \quad d_2 = \frac{D_2}{c}, \quad \text{etc.}$$

or

$$\frac{D_1}{D_2} = \frac{d_1}{d_2}, \quad \text{etc.} \tag{27}$$

The springs should be disposed concentrically with a clearance δ whose value is adequate for the correct centring and allows for change of spring diameters in compression.

Let us assume

$$\delta = \frac{d_1 - d_2}{2}$$

where d_1 and d_2 = wire diameters in adjacent springs.

Here, the smaller-spring diameter D_2 is equal to the larger spring diameter D_1 reduced by an amount

$$2\left(\frac{d_1 + d_2}{2} + \frac{d_1 - d_2}{2}\right) = 2d_1$$

i.e. $D_2 = D_1 - 2d_1$.

Presented in a different notation,

$$\frac{D_2}{D_1} = 1 - 2\frac{d_1}{D_1} = 1 - \frac{2}{c} \tag{28}$$

Joining expressions (26) and (28) together, we shall have:

$$\frac{d_1}{d_2} = \frac{D_1}{D_2} = \frac{i_2}{i_1} = \frac{1}{1 - 2/c}$$

The ratio $\frac{P_1}{P_2}$ of the loads supported by each spring will be

$$\frac{P_1}{P_2} = \frac{d_1^3 D_2}{d_2^3 D_1} = \left(\frac{d_1}{d_2}\right)^2 = \left(\frac{1}{1 - 2/c}\right)^2 \tag{29}$$

Let us compare the deflection of a spring with diameter D_0, supporting load P_0 (see Fig. 365, *I*), with that of an equivalent system comprising two springs supporting the same load $P_0 = P_1 + P_2$. Assume diameter D_1 of the outer spring to be equal to D_0. This spring supports load $P_1 = P_0 - P_2$, whence

$$\frac{P_0}{P_1} = 1 + \frac{P_2}{P_1} = 1 + \left(\frac{d_2}{d_1}\right)^2 = 1 + (1 - 2/c)^2 \tag{30}$$

Deflection λ_1 of the outer spring (which is, at the same time, the deflection of the whole double-spring system) is related to deflection λ_0 in the following way:

$$\frac{\lambda_1}{\lambda_0}=\frac{P_1}{P_0}\frac{i_1}{i_0}\left(\frac{D_1}{D_0}\right)^3\left(\frac{d_0}{d_1}\right)^4 \tag{31}$$

Because

$$D_1=D_0 \quad \text{and} \quad \frac{i_1}{i_0}=\frac{d_0}{d_1}, \quad \frac{\lambda_1}{\lambda_0}=\frac{P_1}{P_0}\left(\frac{d_0}{d_1}\right)^5$$

it follows that (see equation 18)

$$\frac{d_0}{d_1}=\left(\frac{P_0}{P_1}\right)^{1/3}$$

Hence

$$\frac{\lambda_1}{\lambda_0}=\left(\frac{P_0}{P_1}\right)^{2/3}$$

or, substituting $\frac{P_0}{P_1}$ as expressed by equation (30)

$$\frac{\lambda_1}{\lambda_0}=[1+(1-2/c)^2]^{2/3}$$

Given below are the values of ratios $\frac{\lambda_1}{\lambda_0}$ corresponding to the commonly used values of spring index $c = 8$ to 15:

c	8	10	12	15
$\frac{\lambda_1}{\lambda_0}$	1.32	1.38	1.43	1.45

Thus, a system of two concentrically-disposed springs proves to have a 32 to 45% higher deflection capacity than a single-spring system. With a three-spring system, a still greater gain in deflection capacity can be obtained.

Let us see now how the load capacity of a multi-spring system is increased in comparison with a single-spring system at equal deflections in both systems. Assume that the outer spring of the multi-spring system has the same parameters as the equivalent single spring and supports the same load P_0.

From the above assumption that the maximum stresses and working deflections in the springs of a multi-spring system are equal, it follows that the second spring of the smaller diameter, placed within the outer spring, will exert a force expressed by equation (29) as

$$P_1=P_0(1-2/c)^2$$

The system's total load capacity

$$\sum P=P_0+P_1=P_0[1+(1-2/c)^2]$$

The values of ratios $\frac{\sum P}{P_0}$ for different values of c are:

c	8	10	12	15
$\frac{\Sigma P}{P_0}$	1.56	1.64	1.70	1.75

Hence, the load capacity of a double-spring system is from 56 to 75% higher than that of a single equivalent spring.

A third spring located inside the second spring will exert a force

$$P_2 = P_1 (1 - 2/c)^2 = P_0 (1 - 2/c)^4$$

A three-spring system's total load capacity is

$$\sum P = P_0 + P_1 + P_2 = P_0 [1 + (1 - 2/c)^2 + (1 - 2/c)^4]$$

The values of ratios $\frac{\sum P}{P_0}$ for different values of c are:

c	8	10	12	15
$\frac{\Sigma P}{P_0}$	1.88	2.05	2.19	2.25

Thus, a three-spring system will have a 88 to 125% higher load capacity than a single equivalent spring.

The adjacent springs in a concentric-spring system are always made with the coils of opposite hands in order that load may be applied more uniformly and coils of one spring, if broken, may not get caught in clearances between coils of the adjacent spring.

For some large-deflection applications, use is made of concentrically-disposed springs compressed in succession (Fig. 365, *IV*). The total deflection of the system is here equal to the sum of the deflections of the progressively loaded springs. The parameters of these springs should be selected with regard to the conditions expressed by relationships (25)-(27).

Figure 366 shows another type of multi-spring system. Here, a single-wire coiled spring is replaced by a set of identical small-diameter springs disposed around the periphery of the spring caps.

The load and deflection capacities in such a system relate to those of a single equivalent spring differently as compared with a concentric-spring system.

Diameter D_1 of springs that can be accommodated within the diametral envelope taken by an equivalent spring with a diameter D_0 can be found from an approximate relationship

$$D_1 = \frac{0.8\pi (D_0 - D_1)}{n}$$

where n = number of springs; 0.8 = factor allowing for spring centring conditions.

From the relationship it follows that

$$\frac{D_1}{D_0}=\frac{1}{1+0.4n} \qquad (32)$$

Let us compare deflections in a multi-spring and a single-spring systems loaded with an identical force P_0.

With equal forces P_0, the following relationship between the deflections of a multi-spring and a single-spring system can be derived from equation (31):

$$\frac{\lambda'}{\lambda_0}=\left(\frac{D_1}{D_0}\right)^3\frac{i}{i_0}\left(\frac{d_0}{d_1}\right)^4$$

where d_0 and d_1 = wire diameters in single-spring and multi-spring systems.

Using equation (32) and taking, as before, $\frac{i_1}{i_0}=\frac{d_0}{d_1}$, we shall have

$$\frac{\lambda'}{\lambda_0}=\left(\frac{n}{1+0.4n}\right)^3\left(\frac{d_0}{d_1}\right)^5 \qquad (33)$$

The ratio d_0/d_1 will be found from the appropriate equations given before. With identical loads on both systems ($P_0=nP_1$),

$$\frac{d_0}{d_1}=\left(\frac{k_0}{k_1}\right)^{1/3}\left(\frac{n}{1+0.4n}\right)^{1/3}$$

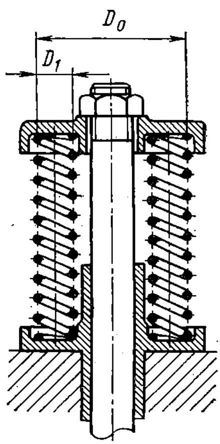


Fig. 366. Multi-spring device

or, as the value of $(k_0/k_1)^{1/3}$ is close to unity,

$$\frac{d_0}{d_1}\approx\left(\frac{n}{1+0.4n}\right)^{1/3}$$

Substituting this relationship into equation (33), we obtain

$$\frac{\lambda'}{\lambda_0}\approx\frac{n^{5/3}}{(1+0.4n)^{4/5}}$$

The values of λ'/λ_0 for different numbers of springs n in a multi-spring system are:

n	6	8	10	12
λ'/λ_0	0.50	0.43	0.37	0.31

It is apparent that the rigidity of a multi-spring system of this type is 2 to 3 times higher than that of a single equivalent spring, the increase in rigidity being proportional to the number of springs.

Let us now compare the load capacity of a multi-spring system and a single-spring system at an identical deflection.

If the systems' deflections are equal, then

$$\frac{d_1}{d_0}=\frac{1}{(1+0.4n)^{3/5}} \tag{34}$$

The load-capacity ratio of the systems (with the values of τ_{al} being equal)

$$\frac{P_1}{P_0}=n\left(\frac{d_1}{d_0}\right)^3\frac{D_0}{D_1}\frac{k_0}{k_1}=n(1+0.4n)\left(\frac{d_1}{d_0}\right)^3\frac{k_0}{k_1}$$

or, with regard to equation (34)

$$\frac{P_1}{P_0}=\frac{n}{(1+0.4n)^{4/5}}\frac{k_0}{k_1}\approx\frac{n}{(1+0.4n)^{4/5}}$$

The values of ratios P_1/P_0 for different spring numbers are:

n	6	8	10	12
P_1/P_0	2.30	2.50	2.65	2.90

As is seen, a multi-spring system has 2 to 3 times the load capacity of a single-spring system.

Thus, the main feature of multi-spring systems of this type is higher load capacity and higher rigidity. Such systems are often used

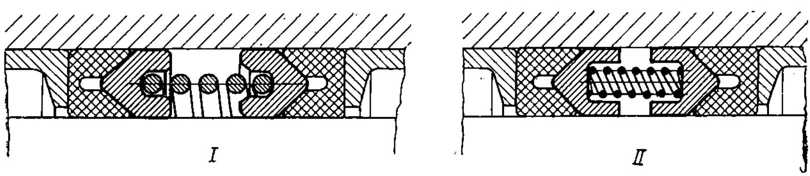


Fig. 367. Lip packings with a single spring (*I*) and multiple springs (*II*)

in spring devices of considerable radial dimensions, for which a single spring proves excessively compliant. In addition, a multi-spring system ensures more uniform distribution of load around the periphery of loaded members.

Figure 367 shows an advantageous application of several small-diameter compression springs in place of a single spring in a lip-type seal of a large diameter.

4.4.8. Conical Compression Springs

Conical compression springs are used most frequently where a variable (i.e. curvilinear) deflection-load relationship needs to be obtained.

The main parameters of conical springs are the angle θ of inclination of the coil-section centre line to the spring axis (Fig. 368) and the variation of the coil pitch along the axis. With a constant pitch t,

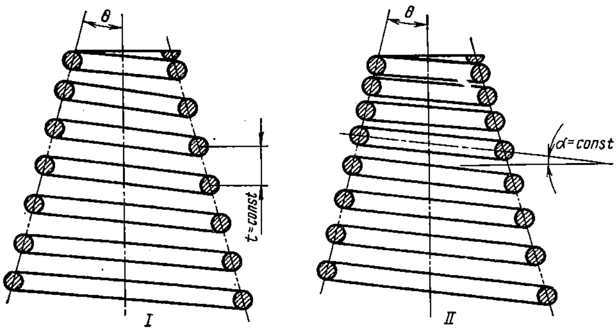


Fig. 368. Conical compression springs
I—with a constant pitch; II—with a constant lead angle

the projection of the spring coils to a plane perpendicular to the spring axis is an Archimedean spiral, whose equation in polar coordinates is

$$R = a\varphi$$

where φ = polar angle; a = constant ($a = t \tan \theta/\pi$).

For a variable pitch, the coil-projection equation takes a more complex form.

A higher spring rigidity in compression is here accounted for by different compliance of coils depending on their diameter. The largest-diameter coils are first to deflect under load. The first end coil rests on the supporting plane; this coil, in turn, supports the second coil, etc. As the coils become closed, their deflection ceases. The rigidity rises gradually as the compression is in progress because the number of free coils decreases and smaller-diameter coils come into operation.

The deflection-load relationship of a conical spring is rendered curvilinear only for a certain combination of spring parameters; it is kept straight on condition that coil pitch t varies directly with deflection λ: $t = \text{const}\ \lambda$.

All other factors being equal, the deflection of coils varies according to equation (19) as the third power of the mean coil diameter: $t = \text{const}\ D^3$.

The mean coil diameter of a conical spring changes along the spring axis in accordance with the formula $D = 2L\tan\theta$, where θ is half the cone included angle and L is the distance between the given coil and the cone apex.

Since $t = \pi D \tan \alpha$, where α is the coil lead angle, then $\tan \alpha = \text{const } L^2 \tan^2\theta = \text{const } L^2$, i.e. with a conical spring, the tangent of the lead angle is bound to decrease towards the spring apex as a square of the distance from this apex.

Used in practice more commonly are springs either with a constant pitch (Fig. 368, *I*) or with a constant lead angle (Fig. 368, *II*); the first type has the lead angle, and the second, the pitch, varying inversely with the distance from the cone apex. The deflection-load

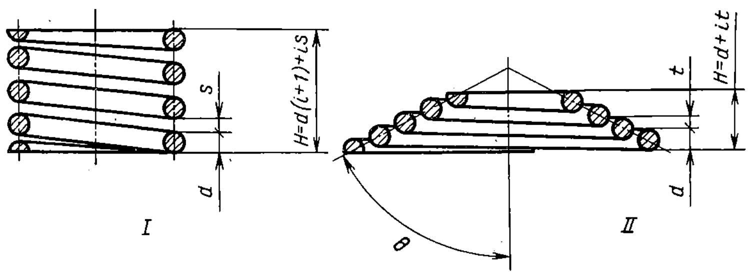


Fig. 369. Free height of compression springs
I—cylindrical spring; *II*—conical spring

relationships in both types are curvilinear; the curves are steeper in the constant-pitch springs and more slightly inclined in the constant-lead-angle springs.

Some other laws may be used for varying the pitch and the lead angle in a spring to obtain a desired type of deflection-load relationship.

Because conical springs have a spiral form in a top plane, they are more difficult to centre than cylindrical springs. For adequate centring, the end coils are uncoiled to a circular form, the concentricity of the coil circles with the spring axis being secured in the process of manufacture.

A special variety of conical springs are telescopic springs, whose coils slide one inside another to form, on complete compression, a flat spiral. Such springs are convenient to use in axially confined spaces.

The height of a completely compressed telescopic spring is equal to its wire diameter d regardless of the number of coils, whereas the solid height of a cylindrical spring is $d(i + 1)$, where i is the number of coils.

The free height of a telescopic spring is $H = d + \lambda$ (Fig. 369). Here, λ is the whole deflection of spring ($\lambda = it$, where t is coil pitch).

In contrast, the free height of a cylindrical spring is $H = d(i + 1) + \lambda$. Here, $\lambda = is$, where s is the amount of clearance between coils.

The main geometric relationship of telescopic springs may be presented in the following form:

$$\tan\theta > d/t$$

where θ = half the cone included angle.

Telescopic springs are centred similarly to all conical springs.

For confined-space high-load applications, use is made of telescopic springs with rectangular-section coils (Fig. 370). Such springs are produced by coiling a steel strip into a flat spiral and subsequently deforming the spiral to a cone. To prevent the development of harmful stresses in the course of deformation, the spiral is annealed before

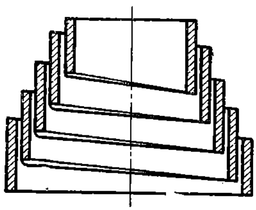


Fig. 370. Rectangular-wire telescopic spring

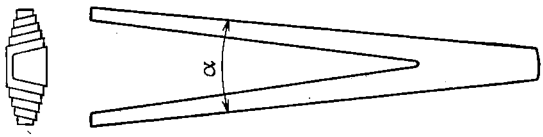


Fig. 371. Double conical spring (spring blank is shown to the right)

or after the process. The end coils (commonly, the larger-diameter lower coil) are uncoiled to a circle and ground flat. Thereafter, the usual heat-treating operations, i.e., hardening and medium-tempering, follow.

In some cases, double conical springs are employed (Fig. 371); these are produced by coiling V-shaped blanks with included angle α equal to the coil lead angle in the spring.

4.4.9. Rectangular Springs

For special applications, use is made of coiled springs having a rectangular form in a top plane (Fig. 372, *I*), which are produced by coiling on special mandrels. A typical application of such springs is the feed spring in the cartridge magazines of firearms. Rectangular springs are unsteady and easily distorted under load. For this reason, they are always mounted in rigid inner or outer guides whose configuration in a top plane corresponds to that of the spring. In some instances, rectangular springs are made as a set of flexible leaves fastened together at the ends (Fig. 372, *II*).

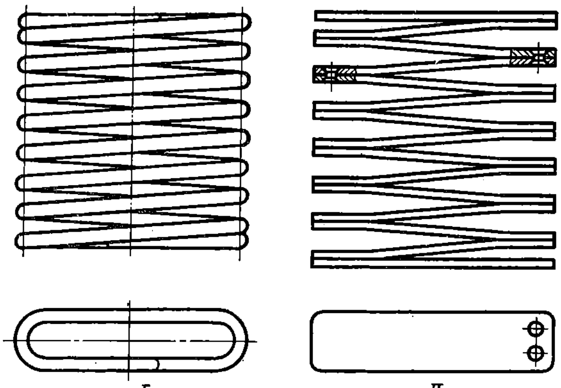


Fig. 372. Rectangular springs

4.5. Helical Extension Springs

Extension springs almost always have their coils wound tightly or even with some initial tension, which is achieved through a displacement of the wire feeder with respect to the coils being wound.

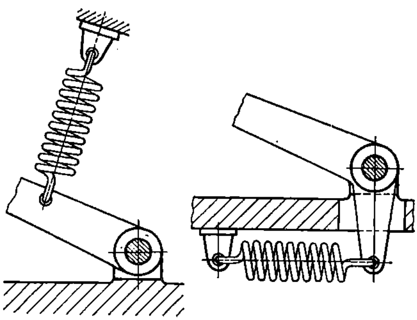


Fig. 373. Some applications of extension springs

The spring ends are made in the form of hooks or loops that connect the spring with the parts it draws to each other. Contrary to compression springs, whose ends need to be guided positively, extension springs operate without guidance, being centred by their supporting (suspension) points only. The suspension from hooks works as an articulation, whereby the spring in tension is capable of varying its position in space within considerable limits. This makes extension springs particularly suitable for connecting parts whose angular position changes in operation, e.g. levers (Fig. 373, *I*, *II*).

The suspension from hooks, however, has some shortcomings. Because of the hooks, an extension spring is always longer than a compression spring of the same deflection capacity. The hooks do not provide adequate means for applying the load centrally; the spring is subject to additional bending loads, and bending stresses arise in the hooks proper, which can in time result in permanent set.

Owing to deformations in the hooks and in the portions where the latter join the helix, the spring elongates and loses its elastic properties. Extension springs can function without the loss of elastisity at lower design stresses only.

For these reasons, extension springs are hardly ever used in vital mechanisms that operate at cyclic loading. In such applications,

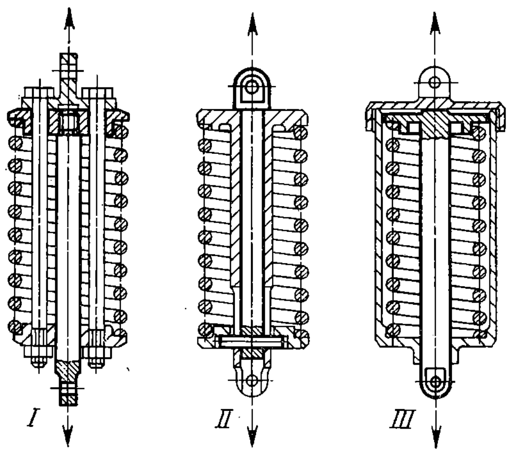


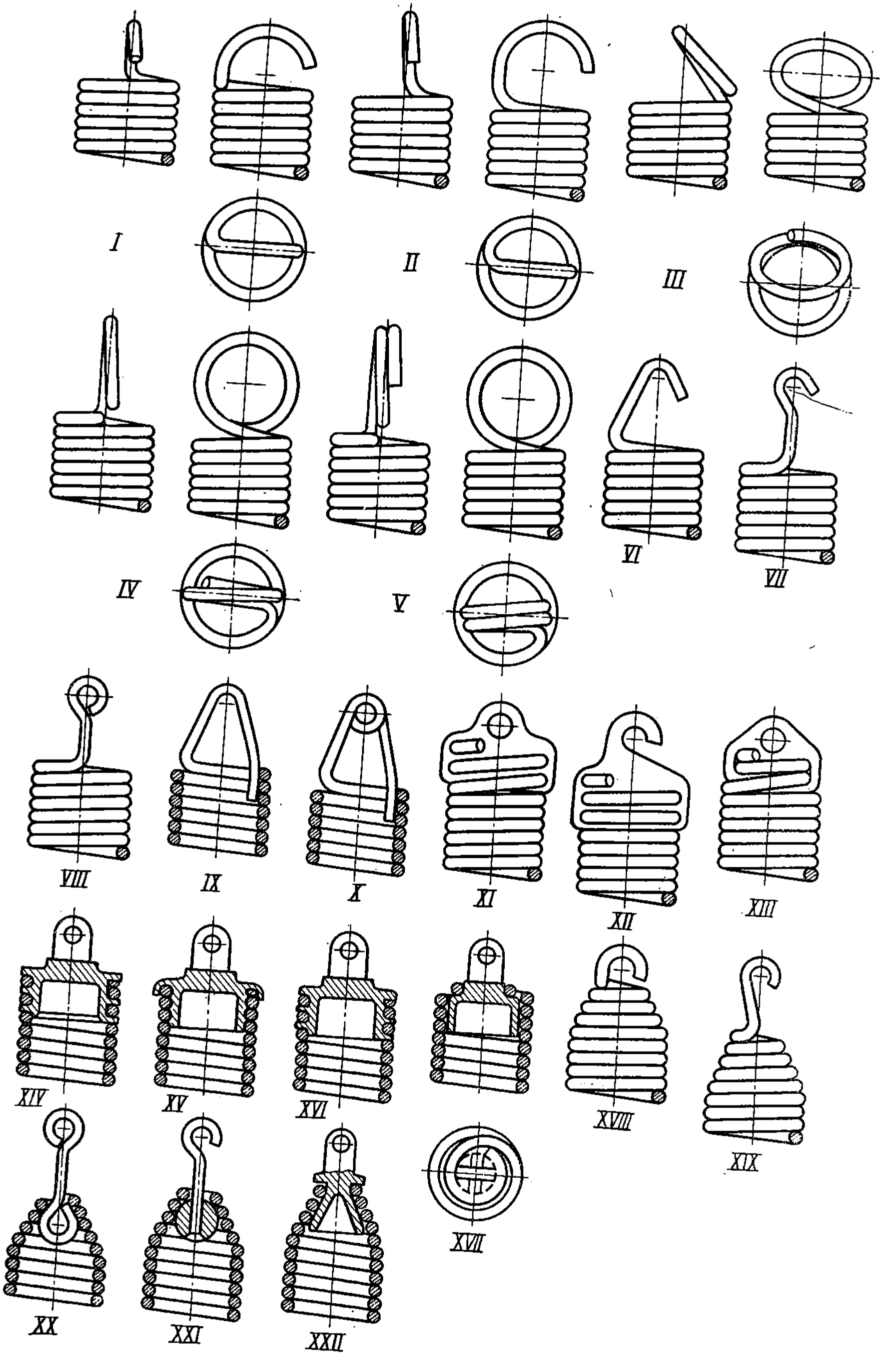
Fig. 374. Compression springs with reversing members for taking tensile loads

compression springs offer smaller overall dimensions and higher reliability.

In applications where an elastic member in a device is to expand, changing its position in space, use is often made of compression springs with reversing members (Fig. 374, *I*, *II*, *III*). The springs of this type, however, are not suited to mechanisms of high-frequency cyclic action because the mass of the reversing members gives rise to additional inertial loads.

Some forms of extension-spring ends are shown in Fig. 375. The most simple are an open half-loop, or hook (Fig. 375, *I*, *II*), and a fully closed loop, which can consist of one coil (Fig. 375, *III*, *IV*), one coil and a half, or two coils (Fig. 375, *V*); such loops are subject to bending stresses and, therefore, are used in springs designed for light service. Also subject to bending stresses are hook-type ends (Fig. 375, *VI*-*VIII*); in addition they are more difficult to make. Triangular loops with the end brought inside the spring helix (Fig. 375, *IX*, *X*) are somewhat stronger.

Light springs of a small-diameter wire are secured in plates having holes through which coils are passed (Fig. 375, *XI*-*XIII*). Here, spring coils must be prevented from going out of the holes; the ne-


I
II
III
IV
V
VI
VII
VIII
IX
X
XI
XII
XIII
XIV
XV
XVI
XVII
XVIII
XIX
XX
XXI
XXII

cessary alignment of the plate with the spring axis is difficult to ensure.

The spring end coils can be turned onto screw-threaded eye plugs and secured thereon by spinning over the plug shoulder (Fig. 375, *XV*) or upsetting the threads (Fig. 375, *XVI*) of the plug. In the end members of this type, the spring coil that comes out of the plug functions under extremely unfavourable conditions, namely, it tends to break off. This condition cannot be eliminated even though the depth of the last groove in the plug is reduced to zero or the plug thread is shaped to a taper.

A similar condition occurs in a plain eye plug exerting the load on the last coil wound up into a small-diameter ring (Fig. 375, *XVII*).

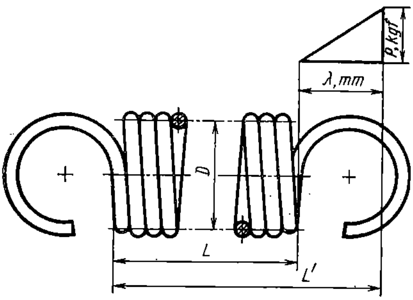


Fig. 376. Deflection-load relationship for extension spring

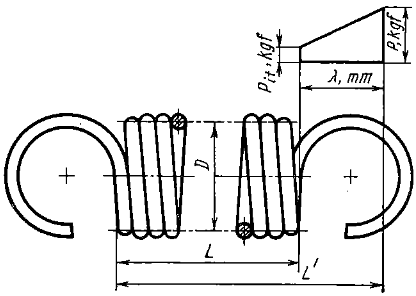


Fig. 377. Deflection-load relationship for extension spring made with initial tension

The most uniform application of load to spring coils is achieved by shaping the end coils like a cone and bending the last coil so as to form a hook (Fig. 375, *XVIII*, *XIX*), or by using swivel hooks and eyes (Fig. 375, *XX-XXII*). However, the manufacture of such springs, particularly those with swivel hooks, where the spring ends have to be wound to a cone with the hooks in place, is difficult.

Among the designs presented in Fig. 375, that with a conical swivel eye (Fig. 375, *XXII*) offers the highest strength. With regard to elastic deformations in end coils, the cone of the swivel eye should be made with a smaller included angle than the inside cone of the coils.

Extension springs are calculated by the same formulas as compression springs. The allowance for bending stresses in hooks and coils (when the load is applied out of centre) is made through a reduction of allowable stresses by 17 to 35% as compared with those for centrally-loaded compression springs.

Figure 376 shows the deflection-load relationship of a conventional extension spring, and Fig. 377, the deflection-load relationship of a spring having an initial tension.

The working length of an extension spring is determined by the formula

$$L = d\,(i + 1)$$

where i = number of active coils.

The working length of the loaded (expanded) spring is

$$L' = L + \lambda$$

where λ = spring deflection.

The developed length of an extension spring

$$L_d = \frac{i\pi D}{\cos\alpha} + L_h$$

where α = lead angle of coils ($\tan\alpha = d/\pi D$); L_h = developed length of hooks.

Approximately,

$$L_d = i\pi D + L_h$$

Extension springs are normally mounted with a preload ensuring that the spring-pressed part will bear against a stop in the initial position. The magnitude of the preload is determined by operating conditions. To allow for probable elongation of hooks in service, the pitch of coils in the preloaded condition is made not less than 1.5 to 2 times the wire diameter.

In an expanded spring, the wire diameter is somewhat reduced because of the increased coil lead angle.

4.6. Helical Torsion Springs

Torsion springs are used to take a twisting, or torsional, moment applied to an end face of the spring. Under the effect of the moment the spring coils are subjected to bending in the plane wherein the moment acts and, in a much lesser degree, to torsion, whose effect is neglected. Torsional moments, both working and reactive, are applied to the spring ends (Fig. 378). Torsion springs operate more steadily if the working moment tends to wind up the spring (Fig. 378, *II*) rather than unwind it (Fig. 378, *I*). The hand of coils and the arrangement of ends should be selected accordingly. For a torsional moment directed clockwise (looking at the spring face), the left-hand coiling should be used, and vice versa.

Torsion springs are wound tightly or with a small clearance between coils to avoid their friction and allow for elongation of spring in torsion.

Figure 379, *I-XVIII* illustrates the common forms of torsion spring ends.

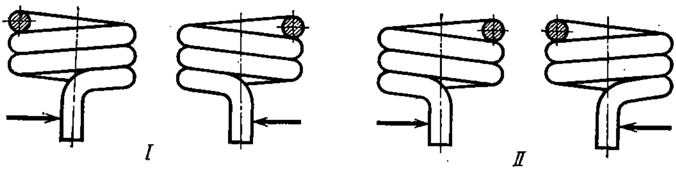


Fig. 378. Hand of coils and disposition of ends in helical torsion springs

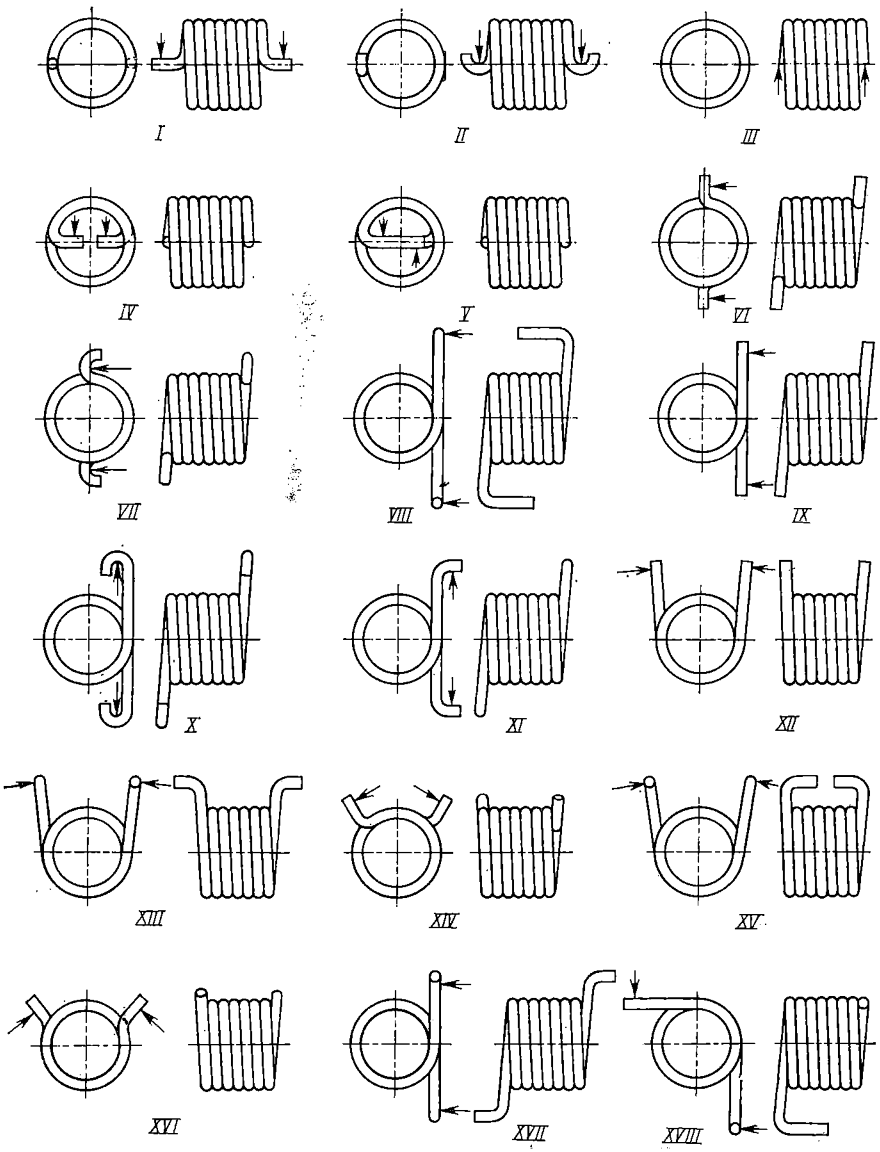


Fig. 379. Arrangement of ends in helical torsion springs

A torsion spring needs to be centred. Centring along the whole length (Fig. 380, *I*) is undesirable because the coil diameter diminishes in torsion (if the above rule for the hand of coiling is observed), and the coils will close on the centring stud. The best solution is to centre the extreme coils (Fig. 380, *II*) at a length not shorter than one and a half or two wire diameters. When centred closely, the end coils should be regarded as inactive.

The diameter-decreasing property of a torsion spring can be used for imparting to the spring a variable stiffness through the introduction of specially-shaped centring elements. In the design shown in

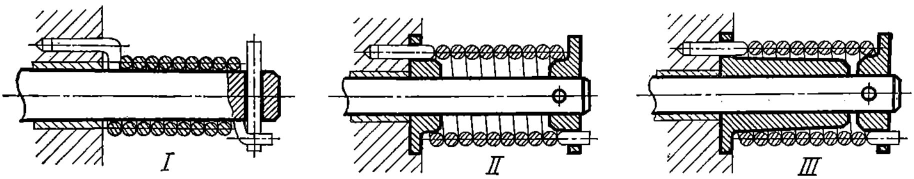


Fig. 380. Centring torsion springs
I—incorrectly; *II*—correctly; *III*—with tapered sleeve

Fig. 380, *III*, the centring sleeve is shaped to a taper. As the spring is winding up, its coils become closed on the taper in succession; thereby, the rigidity of the spring increases with the increased angular deflection.

Helical torsion springs are calculated by the following formulas. The maximum allowable bending stress in spring coils is

$$\sigma = \frac{kM}{W_b}$$

where M = torsional moment; W_b = section modulus for spring wire.

For round wire with diameter d

$$\sigma = \frac{kM \cdot 32}{\pi d^3} \approx 10\frac{kM}{d^3} \tag{35}$$

For square-section wire with square side a

$$\sigma = 6\frac{kM}{a^3} \tag{36}$$

Stress-correction factor

$$k = \frac{4c-1}{4c-4}$$

where c = spring index ($c = D/d = D/a$).

A graph showing the relationship between k and c is presented in Fig. 381.

The allowable bending stresses are from 20 to 30% higher than torsional stresses in compression springs. On the average, $\sigma_{al} = 5{,}000$ to 7,500 kgf/cm².

The spring angular deflection is found from the following equations:

for round wire

$$\varphi = \frac{M \cdot 64Di}{Ed^4} = \frac{2\pi Di}{Edk}\sigma, \quad \text{radians} \qquad (37)$$

$$\varphi = 360^\circ \frac{Di}{Edk}\sigma, \quad \text{degrees of arc} \qquad (38)$$

where E = modulus of elasticity in tension; i = number of active coils;

for square-section wire

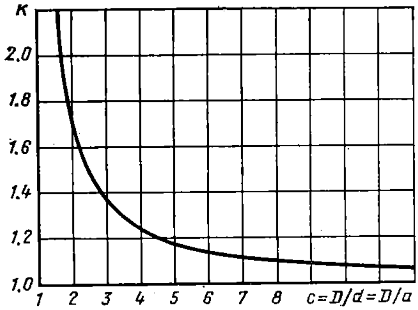


Fig. 381. Relationship between stress-correction factor and spring index for torsion springs

$$\varphi = \frac{M}{E} \cdot \frac{12\pi Di}{a^4} = \frac{2\pi Di}{Eak}\sigma, \quad \text{radians} \qquad (39)$$

$$\varphi = 360^\circ \frac{Di}{Eak}\sigma, \quad \text{degrees of arc} \qquad (40)$$

The above formulas do not take into account the elastic deformation of spring ends.

The maximum allowable angular deflection determined by the steadiness of a spring is

$$\varphi = 120^\circ \sqrt[4]{i}, \quad \text{degrees}$$

The potential energy stored up by a spring in torsion

$$U = \frac{M\varphi}{2}$$

where φ = angular deflection in radians.

The length of the working portion of a spring in the free condition (without the length of ends) is

$$L = d(i+1) + \delta i$$

where δ = clearance between coils.

The elongation of the spring in torsion

$$\Delta L = \varphi \frac{D}{2} \sin \alpha$$

where α = lead angle of coils found by the formula $\tan \alpha = \frac{d+\delta}{\pi D}$ (δ = clearance between coils); φ = deflection angle, radians.

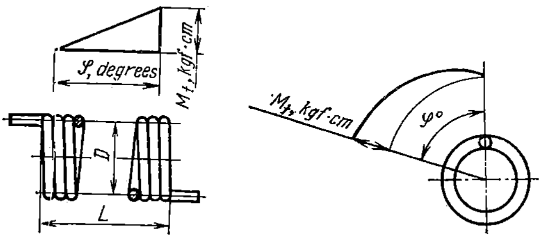


Fig. 382. Representation of deflection-load relationship for torsion springs

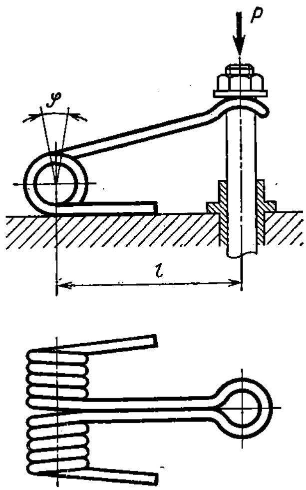


Fig. 383. Torsion spring designed to take transverse load

The increase in coil number during torsion

$$\Delta i = \frac{\varphi}{2\pi}$$

The reduction of wire diameter in torsion is determined on condition that the wire lengths before and after deflection are equal:

$$L = i\frac{\pi D}{\cos \alpha} = \frac{\pi D'}{\cos \alpha}(i + \Delta i)$$

where D' = spring diameter after deflection.

Whence,

$$D' = \frac{D}{1 + \frac{\Delta i}{i}} = \frac{D}{1 + \frac{\varphi}{2\pi i}}$$

The developed length of the spring

$$L_d = \frac{i\pi D}{\cos \alpha} + l$$

where l = developed length of ends.

Approximately,

$$L_d = i\pi D + l$$

Two ways of presentation of the deflection-load relationship for a torsion spring are illustrated in Fig. 382.

Torsion springs are often loaded with transverse forces (Fig. 383). Deflection of the working end is found from the following formula:

$$\lambda = \varphi l + \frac{Pl^3}{3EJ}$$

where l = working-end length; J = moment of inertia for end cross section; φ = angular deflection of spring (in radians), determined from formula (37) where, for the case shown in Fig. 383, substitution is made of $M = \frac{Pl}{2}$ and $i = \frac{i_t}{2}$ (i_t = total number of coils for both arms of spring).

4.7. Multi-Wire Springs

Multi-wire helical compression, extension, and torsion springs, well known but rarely used until recently, are wound from a cable that consists of several wires, or strands, twisted together (Fig. 384, *I-V*). The cable ends are welded around to prevent the strands from uncoiling. The twist angle δ (see Fig. 384, *I*) is normally made equal to 20-30° of arc.

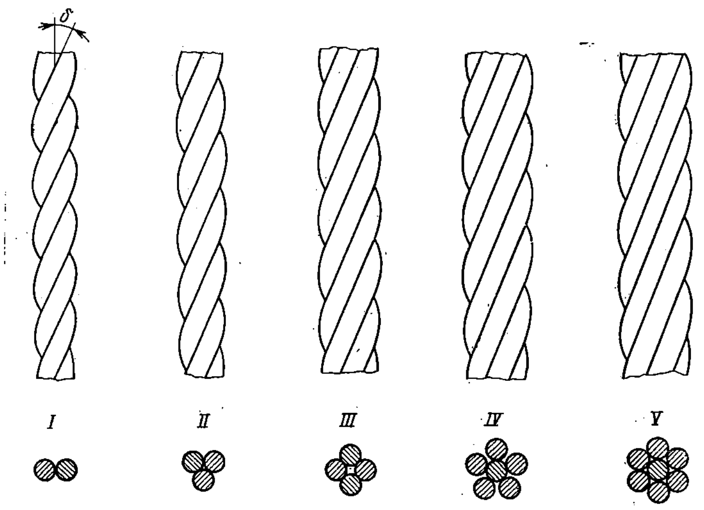


Fig. 384. Multi-strand cables

The cable lay is selected so that during spring deflection the cable twists rather than untwists. The right-hand helix compression springs are produced from cables having the left-hand lay of strands and vice

versa. With extension springs, the cable lay and the hand of the spring-coil helix must coincide. With torsion springs, the lay of strands is of no consequence.

The method of twisting of the cable, its tightness and lead have a major effect on the elastic properties of multi-wire springs. The wires in a twisted cable tend to uncoil. Their mutual position is also altered as a result of winding the spring.

In the free condition of a multi-wire spring, a clearance always takes place between the strands. At the initial stage of loading, each strand of the cable functions independently, and the deflection-load curve is sloping slightly. As the loading is in progress, the cable twists, its strands come close together and begin to work as a whole; the stiffness of the spring rises. For this reason, the deflection-load curve of a multi-wire spring has a break point a corresponding to the initial moment at which the strands become closed (Fig. 385).

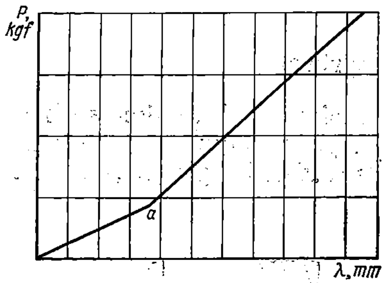


Fig. 385. Deflection-load relationship for multi-wire compression spring

The advantages of multi-wire springs can be seen from the following considerations. The use of several thin wires instead of one larger-diameter wire enables the designer to specify higher allowable working stresses because of a higher strength inherent in thin wires. A coil made up of the small-diameter strands exhibits better elasticity than an equivalent massive coil, in part owing to the higher allowable stresses, but mainly because of a larger value of spring index $c = D/d$ for each strand, which greatly influences the spring rigidity.

The gradually sloping deflection-load curve of multi-wire springs may prove useful in some applications where large deflections need to be obtained in axially- and radially-confined spaces.

Another peculiarity of multi-wire springs is higher oscillation-damping capacity due to the friction between strands during spring deflection. This feature can be used for the dissipation of energy and absorption of shocks and vibrations arising from impact loads; it also makes for self-damping of resonant oscillations in spring coils.

The higher friction between coils, however, causes wear that leads to a decrease in the spring fatigue life.

When estimating the compliance of multi-wire and single-wire springs, comparison is often made erroneously between the springs having identical cross-section areas of coils (with multi-wire springs, the total cross-section area of the strands).

In doing so, the fact is overlooked that the load capacity of multi-wire springs is lower than that of single-wire springs (with all other things being equal) and diminishes with increase in the number of strands.

For this reason, comparative advantages of multi-wire and single-wire springs, and also of multi-wire springs with different numbers of strands can only be determined on the assumption that the springs have identical load capacities. The advantages of multi-wire springs revealed by such an estimate prove to be smaller than could be expected.

Compare the deflection capacities of a multi-wire and a single-wire compression spring having identical mean coil diameters, number of coils, and safety factors, and supporting an identical load P.

Assume roughly that the multi-wire spring consists of a number of springs with small-section coils, working in parallel.

On this assumption, diameter d' of a strand in the multi-wire spring is related to diameter d of a single wire in the following way

$$\frac{d'}{d}=\sqrt[3]{\frac{1}{n}}\sqrt[3]{\frac{\tau}{\tau'}\frac{k}{k'}} \tag{41}$$

where n = number of strands; τ and τ' = allowable shearing stresses; k and k' = stress-correction factors*.

Since the values of $\sqrt[3]{\tau/\tau'}$ and $\sqrt[3]{k/k'}$ are close to unity, equations (41) can be put down as

$$\frac{d'}{d}\approx\sqrt[3]{\frac{1}{n}} \tag{42}$$

The ratio of masses of the springs compared is

$$\frac{G'}{G}=\left(\frac{d'}{d}\right)^2 n$$

or, substituting d'/d from equation (42)

$$\frac{G'}{G}=n^{1/3}$$

The values of ratios d'/d and G'/G as dependent on the number of strands are:

n	2	3	4	5	6	7
d'/d	0.80	0.70	0.63	0.58	0.55	0.53
G'/G	1.25	1.43	1.60	1.70	1.80	1.90

As can be seen from the above data, the reduction in wire diameter in multi-wire springs is not so large as to yield any substantial gains

---

* Index refers to multi-wire spring.

in strength even for small values of d and d' (incidentally, this justifies the above assumption that factor $\sqrt[3]{\tau/\tau'}$ is close to unity).

The ratio of deflection λ' of the multi-wire spring to deflection λ of the single-wire spring is

$$\frac{\lambda'}{\lambda}=\frac{d}{d'}\frac{k}{k'}\frac{\tau'}{\tau}$$

Substituting d'/d from equation (41), we obtain

$$\frac{\lambda'}{\lambda}=n^{1/3}\left(\frac{\tau'}{\tau}\right)^{4/3}\left(\frac{k}{k'}\right)^{4/3}$$

The value of τ'/τ, as indicated before, is close to unity. Therefore

$$\frac{\lambda'}{\lambda}=n^{1/3}\left(\frac{k}{k'}\right)^{4/3}$$

The values of λ'/λ calculated from this equation for different numbers of strands n (with k found at $c = 6$) will be:

n	2	3	4	5	6	7
λ'/λ	1.35	1.57	1.82	2.00	2.15	2.25

It is apparent that with identical loads on the springs, changeover to the multi-wire spring will give, for a reasonable number of strands, a gain of 35 to 125% in deflection capacity.

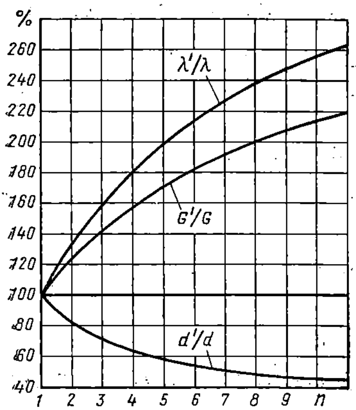


Fig. 386. Ratios of deflections λ'/λ, masses G_1/G, and wire diameters d'/d as functions of number of strands for compression/extension springs of equal load capacity

A composite diagram of change in factors d'/d, λ'/λ and G'/G depending on the number of strands in identically loaded and equally strong multi-wire springs is presented in Fig. 386.

Along with increase in mass due to larger number of strands, increase in the coil-section diameter should also be taken into account. For the number n of strands ranging from 2 to 7, the coil-section diameter is, on the average, 60% larger than the diameter of the equivalent single wire. As a result, the pitch and overall length of the spring need to be increased in order to retain the specified amount of clearance between coils.

The gain in deflection capacity offered by multi-wire springs can be obtained quite readily in a single-wire spring. For this purpose,

it is necessary to increase spring coil diameter D, reduce wire diameter d, and raise the allowable working stress (through the use of high-grade spring steels). In the end, a single-wire spring that compares to a multi-wire spring in deflection capacity, will have a lower mass, smaller overall dimensions and, owing to far easier manufacture, will be considerably cheaper than the corresponding multi-wire spring. Additional shortcomings of multi-wire springs are:

(1) in compression springs, it is impossible to form the ends correctly (by grinding them flat) in order to achieve the central application of load; the load is always applied out of centre to some extent, which results in additional bending of the spring;

(2) the springs are difficult to manufacture;

(3) owing to manufacturing conditions, the spring characteristics vary; they are unstable and show poor repeatability;

(4) as a result of friction, the strands are subject to wear in repeated working cycles, which leads to a rapid decrease in the spring fatigue life. The latter drawback excludes the use of multi-wire springs in applications at durable cyclic loading.

Multi-wire springs are applicable for static and variable-dynamic loading with a limited total number of cycles.

4.8. Annular Springs

Annular springs consist of a set of alternating rings with outside and inside conical surfaces (Fig. 387, *I*).

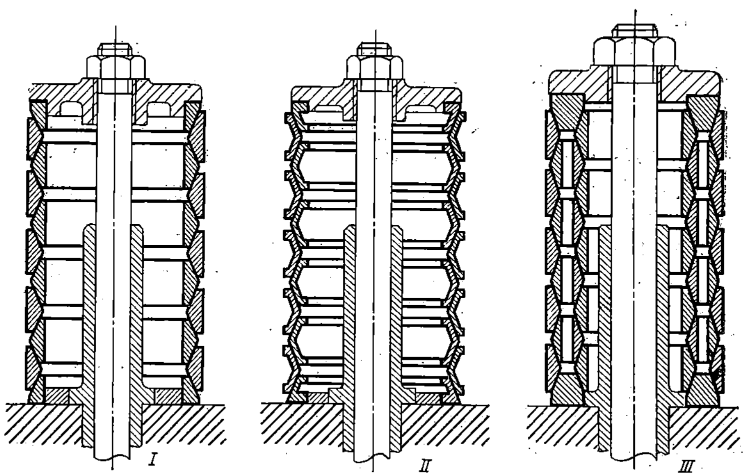


Fig. 387. Annular springs

An axial load applied to such a spring gives rise to high pressures on the conical surfaces, causing expansion of outside rings and con-

traction of inside rings. Owing to a small taper angle, relatively small radial deflections of rings are transformed into considerable axial movements. The total axial movement of all the rings defines the spring deflection.

A V-shape is given to the ring cross-section for increased elasticity (Fig. 387, *II*). An annular spring comprising two sets of concentric rings designed to take high axial loads is illustrated in Fig. 387, *III*.

Half the included angle β of the ring taper (Fig. 388) is made somewhat larger than the angle of friction φ ($\tan \varphi = f$, where f is the coefficient of sliding friction) in order that on reduction or removal of load, the elastic forces produced in the rings by preceding loading might overcome frictional forces and cause the backward movement of the rings, i.e. return of the spring to the original, free condition.

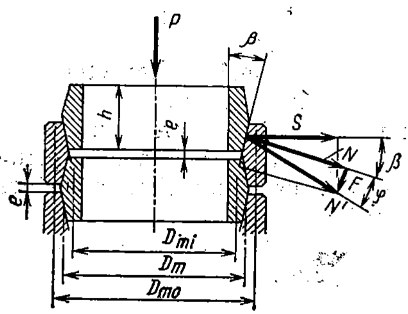


Fig. 388. Forces acting on annular spring

For steel rings, $f = 0.12$ to 0.15 and $\varphi = 7$ to $9°$. In practice, half the included angle β of taper is assumed to be from 12 to 15°.

The frictional forces arising from the movement of the rings determine a high damping capacity of annular springs. About 60% of energy taken by the spring during a loading cycle is transformed into expendable friction work and dissipated in the form of heat into the ambience. An annular spring is virtually a combination of a spring and a damper dash-pot. Annular springs are matchless for service under periodic impact loads where it is essential along with elastic damping to ensure the absorption of impact energy and prevent oscillations in the system.

The use of annular springs is objectionable where a spring is to work as an accumulator to store up energy during the loading cycle and release it in the unloading cycle (the most frequent application of a spring). Here, common helical springs have all advantages because they possess a small elastic hysteresis and are, in terms of energy, accumulators with almost hundred-percent efficiency. Annular springs are mainly employed as compression springs. With the use of reversing members (see Fig. 374), they can also be subjected to tension loading.

Annular springs are much more difficult to produce than helical springs. For maximum strength, the rings should be manufactured from individual blanks by hot stamping (to set up the required flow of grains) and subsequent rolling to size. The mechanical properties

of rings produced by turning from bar or tube are lower because of unfavourable grain flow. After heat treatment, the rings are ground on the conical working surfaces and shot-peened or roller-burnished.

The working stresses for annular springs are tensile stresses in the outside rings and compressive stresses in the inside rings.

Under the action of axial load, forces distributed over the ring circumference arise on the conical surfaces. In Fig. 388, these forces on one of the surfaces are brought conventionally into one point disposed at a diametral section.

Added to frictional force $F = Nf$, force N perpendicular to the taper generatrix deviates at an angle φ whose tangent is equal to $F/N = f$.

The resultant force N' is found on the assumption that the axial components of force N' are equal to acting load P:

$$P = N' \sin(\beta + \varphi)$$

whence

$$N' = \frac{P}{\sin(\beta + \varphi)}$$

Distributed load q from this force, acting along the ring's circumference is

$$q = \frac{P}{\pi D_m \sin(\beta + \varphi)} \tag{43}$$

where D_m = mean diameter of spring.

The outside ring is expanded by distributed forces s, which are radial components of forces q acting on the lower and upper tapers.

Thus, distributed load s_d expanding the ring (and acting like a pressure) equals

$$s_d = 2q' \cos(\beta + \varphi)$$

Substituting q from equation (43), we obtain

$$s_d = \frac{2P}{\pi D_m \tan(\beta + \varphi)}$$

Tensile stress in the outside ring

$$\sigma_t = \frac{s_d D_m}{2F_{or}} = \frac{P}{\pi F_{or} \tan(\beta + \varphi)} \tag{44}$$

where F_{or} = cross-sectional area of outside ring.

Similarly, compressive stress in the inside ring

$$\sigma_c = \frac{P}{\pi F_{ir} \tan(\beta + \varphi)} \tag{45}$$

where F_{ir} = cross-sectional area of inside ring.

The elongation of the outside ring around the circumference

$$\Delta_{cf} = \pi D_{mo} \frac{\sigma_t}{E}$$

where D_{mo} = mean diameter of outside ring; E = modulus of elasticity of ring material.

Increase in the outside-ring diameter

$$\Delta_{od} = \frac{\Delta_{cf}}{\pi} = D_{mo} \frac{\sigma_t}{E}$$

Decrease in the inside-ring diameter

$$\Delta_{id} = D_{mi} \frac{\sigma_c}{E}$$

where D_{mi} = mean diameter of inside ring.

The axial displacement of the outside ring relative to the inside ring

$$\delta = \frac{\Delta_{od} + \Delta_{id}}{2 \tan \beta} = \frac{1}{2E \tan \beta} (D_{mo}\sigma_t + D_{mi}\sigma_c)$$

Substituting σ_t and σ_c from equations (44) and (45), we have

$$\delta = \frac{P}{2\pi E \tan \beta \tan (\beta + \varphi)} \left(\frac{D_{mo}}{F_{or}} + \frac{D_{mi}}{F_{ir}} \right)$$

With equal cross-sectional areas of the outside and inside rings ($F_{or} = F_{ir} = F$), or, to put it differently, at equal absolute values of stresses in the rings ($\sigma_t = \sigma_c = \sigma$)

$$\delta = \frac{P}{2\pi EF \tan \beta \tan (\beta + \varphi)} (D_{mo} + D_{mi}) = \frac{\sigma}{E \tan \beta} \frac{D_{mo} + D_{mi}}{2}$$

$$= \frac{\sigma}{E \tan \beta} D_m$$

where D_m = mean diameter of spring.

Under the action of load P, the total spring deflection will be

$$\lambda = (i - 1)\,\delta = \frac{\sigma (i - 1)}{E \tan \beta} D_m \qquad (46)$$

where i = total number of rings in spring, including end rings.

This formula holds true if the end rings take part in the operation of the spring, elastically contracting under load, and if stresses in the end rings are equal to those in the active rings (for this purpose, the cross section of the end rings should be one-half that of the active rings).

Where the end rings are inside rings tightly fitted in the spring caps (see Fig. 387, *I*) and cannot contract, equation (46) takes the

following form:

$$\lambda = \frac{\sigma}{E \tan \beta} [D_m (i-3) + D_{mo}] \quad (47)$$

The energy absorbed by the spring during one loading-relieving cycle is given by

$$U = \zeta \frac{P\lambda}{2}$$

where ζ = 0.6-0.7 = energy dissipation factor.

The height of rings h (see Fig. 388) is normally made from 3 to 5 b_m, where b_m is the mean thickness of rings. The D_m/b_m ratio usually ranges from 15 to 30.

Axial clearance e between rings (see Fig. 388) must be selected with regard to the spring working deflection. The total of clearances e equal to $(i - 2)\, e$ should be either larger than, or equal to, maximum deflection λ_{max} of the spring if rigid support is to be secured to prevent the rings from an excessive stress at accidental overloading. This condition results in the relationship

$$e \geqslant \frac{\lambda_{max}}{i-2}$$

The total free length of the spring (with single-sided end rings) is

$$L = \frac{i-1}{2} (h+e) + \delta_{max}$$

On the average, the allowable working stresses for annular springs are assumed to be equal to 30-40 kgf/mm^2.

The ring working surfaces need to be properly lubricated. Means for the dissipation of heat arising from friction should be provided for springs of repeated high-frequency action.

4.9. Washer Springs

These springs are used to sustain considerable loads at small deflections.

The most common type of such springs is a Belleville spring washer (Fig. 389, *I*), which is produced by stamping a blank of sheet spring steel to a conical shape.

The Belleville washers are available in the following dimensions: thickness, from 1 to 20 mm; diameter, 30 to 300 mm; d/D ratio, 0.5 to 0.3; and angle θ, 2 to 6°.

The washer faces are ground flat to form annular bearing surfaces. When being loaded centrally, the washer is subject to bending, and its faces get closer together (commonly, within the range of 0.5 to 0.8 f, where f is the height of the washer's frustum of cone).

Close in shape to Belleville washer springs are spherical dish springs (Fig. 389, *II*). Some other varieties of washer springs are presented in Fig. 389, *III-VIII*.

For improved flexibility, washer springs are made waved or of a bellows-type (Fig. 389, *VII*, *VIII*).

Depending on the spring configuration and thickness of the blank, complex-shape washer springs are cold- or hot-stamped and subjected

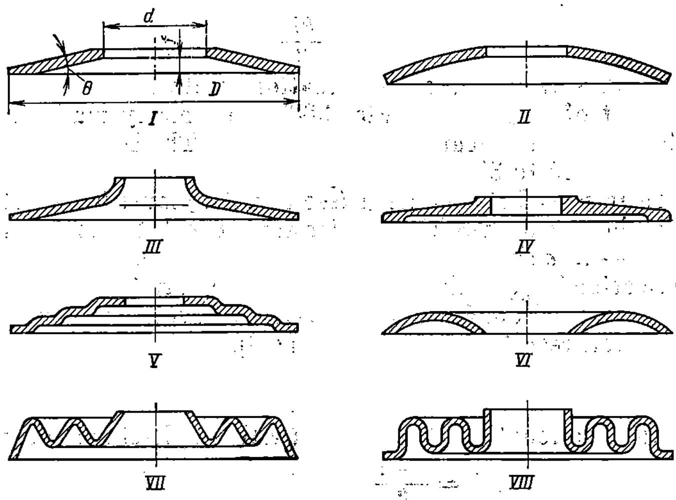


Fig. 389. Washer springs

to the standard heat treatment for a given grade of steel (hardening and medium-tempering).

The strength of spring washers produced by turning from cylindrical blanks is substantially lower than that of stamped washers.

Several washers used in combination form a spring device of increased flexibility (Fig. 390, *I*, *II*, *III*); such combinations are often employed as shock absorbers and dampers for high loads.

Washers with weight-reducing holes (Fig. 390, *IV*), star-like (Fig. 390, *V*), waved and bellows-like washers (Fig. 390, *VI*, *VII*) are close in deflection capacity to helical springs.

Steep-taper Belleville spring washers and large-radius spherical washers are calculated approximately as circular plates supported at the circumference and centrally loaded.

The calculation of wave and other complex-shape washer springs is difficult; the deflection-load relationships and allowable working stresses for such springs are mostly found experimentally.

For some applications in corrosive media, springs are made in the shape of bellows with single- or multi-layer walls (Fig. 390, *VIII*); the materials employed are sheet brass or bronze.

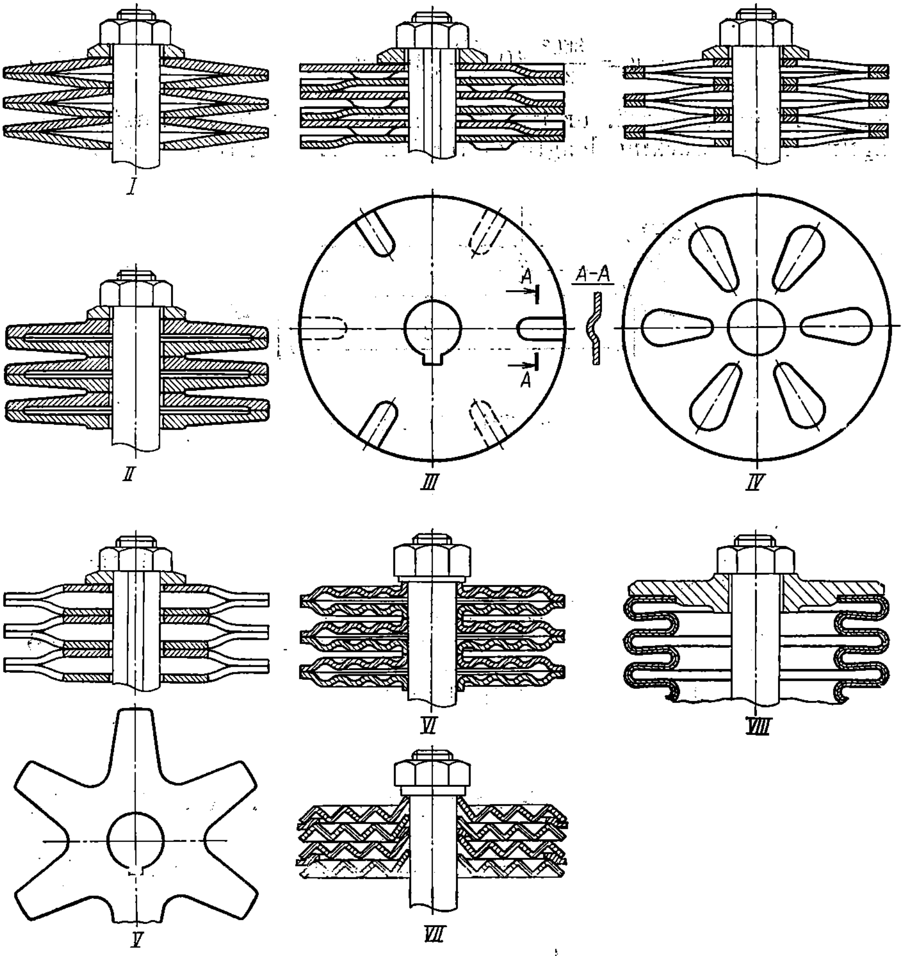


Fig. 390. Types of washer-spring combinations

4.10. Torsion Bars

A torsion bar, i.e. a cylindrical rod rigidly secured at its ends in members that are subjected to a torsional moment causing their angular deflection relative to each other, is the simplest type of a torsion spring. The bar ends are fixed most often with splines. A typical construction of a cylindrical torsion bar is shown in Fig. 391. The angular deflection of the bar is

$$\varphi = \frac{Ml \cdot 32}{G\pi d^4} \approx 10 \frac{Ml}{Gd^4}, \text{ radians} \qquad (48)$$

or

$$\varphi = \frac{360^\circ}{2\pi} 10 \frac{Ml}{Gd^4}, \text{degrees} \tag{49}$$

where M = torsional, or twisting, moment acting on torsion bar kgf/mm; l = working length of torsion bar, mm; G = modulus of

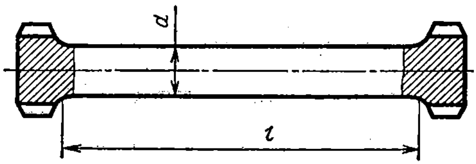


Fig. 391. Torsion bar

elasticity in shear, kgf/mm^2; d = diameter of the working portion of torsion bar.

The maximum torsional shearing stress at the peripheral fibers of bar cross-section

$$\tau = \frac{M}{W} = \frac{M \cdot 16}{\pi d^3} \approx 5 \frac{M}{d^3} \tag{50}$$

The torsional moment sustained by the bar

$$M = 0.2 d^3 \tau \tag{51}$$

Substituting this expression into formula (49), we shall obtain.

$$\varphi = \frac{2l\tau}{Gd}, \text{ radians} \tag{52}$$

The allowable torsional shearing stress τ_{al} in torsion bars made from spring steels is 40 to 60 kgf/mm^2.

As is seen from formula (52), the angular deflection at given τ is wholly determined by l/d ratio.

The use of torsion bars is particularly advantageous for interconnection of aligned hollow shafts. In that case, the torsion bar may have a considerable length, and its angular deflection can reach several tens of degrees.

Owing to simple design and small radial dimensions, torsion bars are extensively used at present in the engineering industry as a means of flexible connection between rotatable parts, e.g. for compensation of torque nonuniformity in piston-type machines. At the same time, torsion bars are a suitable means of making up for the misalignment or skewness of parts connected together. Torsion bars

are also used as substitutes for compression or leaf springs to support transverse loads. For this purpose, one end of the bar is fixed in the housing, and the other is furnished with a lever to which a transverse forse is applied (Fig. 392). Such constructions are used for elastic suspension of automobile wheels, for driving valves in piston-type combustion engines, etc.

The deflection of the lever end along the line of action of the force for the instance shown in Fig. 392 is

$$\lambda = 2R \sin \frac{\varphi}{2}$$

where φ = angular deflection found by equation (49).

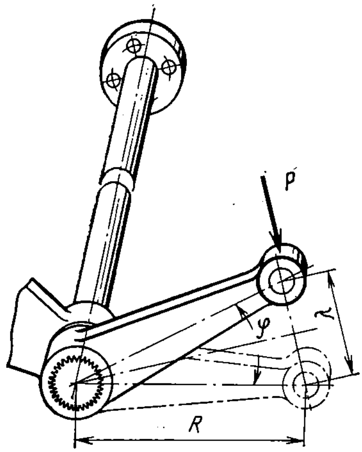


Fig. 392. Torsion bar designed to take transverse load

4.11. Rubber Shock and Vibration Absorbers

Rubber has excellent properties as a shock and vibration absorbing material, i.e. high compliance and great internal friction, the latter providing for effective damping of vibrations. The modulus of

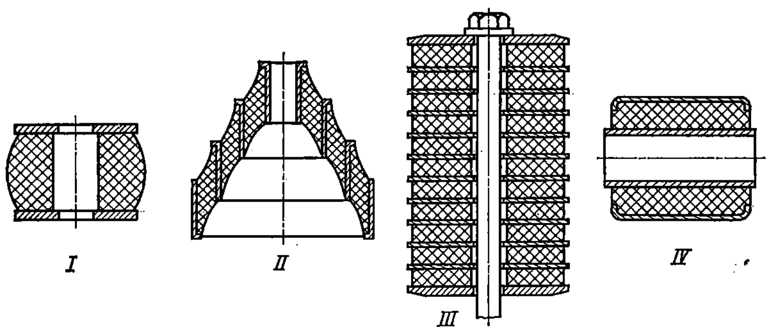


Fig. 393. Rubber shock-and-vibration absorbers

elasticity of rubber is very low (from 0.8 to 1.0 kgf/mm^2), which is about 1/20,000 that of steel. The tensile strength is, on the average, from 1 to 2 kgf/mm^2.

Rubber absorbers are used for elastic suspension of machinery and apparatus, for damping shocks and impacts in the chassis of motor vehicles, etc.

Some designs of rubber absorbers are presented in Fig. 393, *I-IV*. The absorber shown in Fig. 393, *III* consists of a set of rubber discs separated with metal washers; it features a long working movement and high internal friction.

Cylindrical absorbers comprising a rubber sleeve vulcanized to external and internal metal shells (Fig. 393, *IV*) are widely used in transportation engineering. These absorbers, sometimes referred to as silent blocks, are capable of taking both transverse forces and torsion moments. They can replace plain bearings for operation at small angular deflection. Silent blocks are mounted in the spring eyes of car suspensions, on wheel axles, etc.

Tube Connections

5.1. Flexible Couplings

Oil, water, air, and vacuum conduits are made up of steel, brass, duralumin, and plastic tubes and pipes.

Flexible couplings (Fig. 394, *I*, *II*) are used for tubes of 10 to 60 mm in diameter working at small and medium pressures. A piece

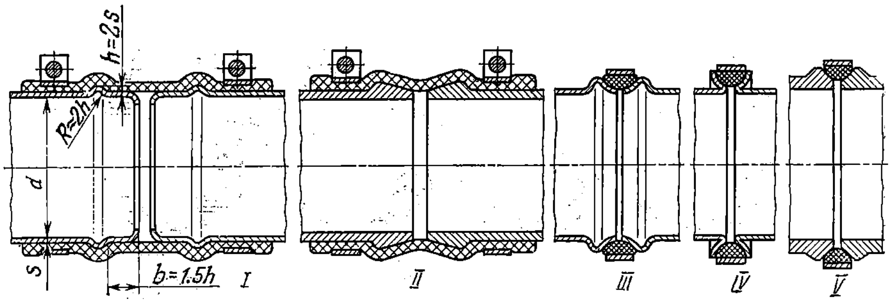


Fig. 394. Flexible couplings

of hose made from rubber, plastics or like materials is fitted over the ends of the tubes being joined and fixed with wire bindings or clamping collars.

The flexibility of the hose allows for some misalignment of the tubes connected. The axial load-carrying capacity of such a coupling is low. Where the joint is to be loaded with axial forces, the tubes must be rigidly fixed in place.

For easier mounting of the hose sleeve, the tube ends are made tapered; they are also provided with annular crimps or ridges (Fig. 394, *I*, *II*), which make for better sealing and holding of the sleeve on the tubes.

Fig. 394, *III*-*V* shows a method of joining two lengths of tube with elastic rings tightened by clamping collars. This joint is used for rigidly fixed tubing; it requires a fairly accurate alignment and small axial clearance between the tubes connected. The tightening gives rise to forces that tend to thrust the tubes away from each other.

The reliability of flexible couplings and convenience of their assembly and disassembly largely depend on the design of clamping collars. Wholly closed collars (Fig. 395, *I*), which are tightened with a screw through block *1* and saddle *2*, bearing on the hose sleeve are used to join axially adjustable pipes and tubes. This type of collar is difficult to mount on and detach from a fixed tube connection.

Fig. 395. Clamping collars

Another drawback to such collars is inadequate tightening of the hose sleeve at the portions where the annular collar band joins the saddle.

In the design shown in Fig. 395, *II*, the band ends are fixed in openings made in adjacent portions of the band. By changing the position of the band ends, the collar can be adjusted to the tube diameter. Moreover, this construction simplifies assembly. Tightening is done by means of a screw turned into block *3* welded to the band. When turned in, the screw, which bears against saddle *4*, stretches the band, pressing it around the hose sleeve. Because the band is guided by guides *m* in the saddle, the tightening is here more uniform than in the foregoing construction.

Open collars with bent-out ends for a screw fastener (Fig. 395, *III* are inconvenient to assemble since here bolts and nuts need to be used. The hose sleeve is not tightened at the collar portion where the bolt is located. The collar cannot be clamped strongly because the band ends will deflect towards each other.

For higher stiffness of the tightening joint, the collar is made channeled by stamping (Fig. 395, *IV*). In the collar illustrated in Fig. 395, *V*, channel-section lugs are welded to the band ends; the clamping screw is turned into the lower lug strengthened with a flat insert *n*. To ensure uniform tightening, one band end is made to overlap the other end (*q*).

Placing and removal of the clamping collar shown in Fig. 395, *VI* are simplified by the use of screw *5* mounted at one end of the collar; the screw pushes block *6* welded to the other end of the collar and projecting through an elongated opening in the first end.

The band ends in the collar presented in Fig. 395, *VII* are provided with loops which hold cylindrical blocks *7* and *8*. To receive a clamping screw, block *7* has a clearance hole, and block *8*, a corresponding threaded hole. When turned in, the screw, bearing with its shoulder against block *7*, draws block *8* toward itself. Because the blocks are free to turn within the loops, the screw is not subject to bending, but is in tension only. Fig. 395, *VIII* shows a similar design for a pressing screw.

In a quick-acting clamping collar (Fig. 395, *IX*), upper block *9* is inserted into cup-type member *10* welded to the collar end. A left-hand screw thread is tapped in the block to receive the clamping screw, whose end is passed through a clearance hole in block *11* and fixed therein against axial movement. To dismantle the joint, it is only necessary to turn the screw out several turns so that the upper block comes out of the cup member and to swing the screw in the direction indicated with the arrow.

A similar construction, which makes use of a pressing screw (Fig. 395, *X*), has swingable clamp *13* pivoted on pin *12* mounted in the collar. The screw, installed at the top of clamp *13*, acts upon block *14* welded to the opposite end of the collar and passed through the opening in clamp *13*.

A mechanism that ensures strong clamping without the use of any tool is shown in Fig. 395, *XI*. Latch *15* pivoting on pin *16* at the upper end of the collar is hinge-jointed to link *17*. The hooked end of link *17* is introduced into hole *s* in the lower end of the collar; by turning latch *15* in the direction of the arrow, the clamp is tightened. Bearing on the upper band, latch *15* passes through the dead point and locks the joint.

In the construction illustrated in Fig. 395, *XII*, clamping is effected by means of screw *18* having a left-hand thread; the screw is mounted in a plain hole of sleeve *19* and fixed axially therein. The

threads of the screw engage spirally directed notches t on the collar band. When rotated, the screw interacts with the notched end of the band as if with a rack, thus tightening the collar. To prevent fast wear of the notches, the band is heat-treated to a hardness not less than Rc 45. With a fairly large number of notches, the collar may be used for clamping tube joints over a large range of diameters.

For complete detachment of the collar, the screw should be rotated until disengaged from the notched end of the band.

5.2. Flanged Fittings

Flanged fittings (Fig. 396) are used mainly for large-diameter tubing. Their application for tubes of small diameter is restricted by

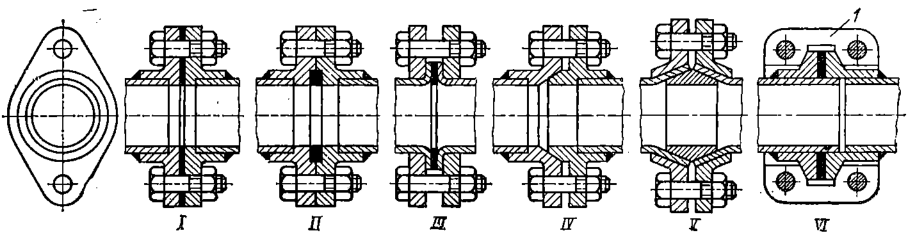


Fig. 396. Flanged fittings

inconvenient assembly of the joint (a large number of fastening points and the need for use of bolts and nuts). Small-diameter tubes are

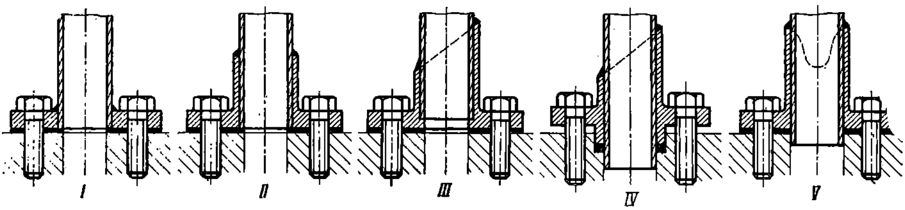


Fig. 397. Joining tubes to housing-type parts through flanged fittings

usually joined with rhomboidal flanges. Flanged joints are packed with sheet gaskets (Fig. 396, *I*) or *O*-rings (Fig. 396, *II*). Fig. 396, *III* illustrates a joint that uses detachable flanges on tubes having flanged ends.

In the design according to Fig. 396, *IV*, the flanged fitting is tightened on conical surfaces, and in the joint shown in Fig. 396, *V* the tubes being connected have flared ends that are tightened against

a double-cone insert. In a conical-flange fitting, the flanges are tightened within conical recesses in outer detachable parts *1*.

Flanged fittings for coupling tubes to housing-type parts have found wider application. The flanges are joined to tubes by soldering or welding (Fig. 397, *I*). For higher strength and rigidity, flanges are provided with sleeves (Fig. 397, *II*). The strength of the weld is improved by making the sleeve end slanted (Fig. 397, *III*, *IV*) or complex-shaped (Fig. 397, *V*).

5.3. Union-Type Fittings

Tubes of small diameters (5 to 20 mm) and walls of 0.2 to 0.5 mm thick are most often interconnected with union-type fittings. Such fittings withstand pressures up to 30-50 kgf/cm^2; they are small and convenient in assembly.

For connection of tubing made from solderable and weldable materials, use is made of three-piece flareless-tube fittings (Fig. 398).

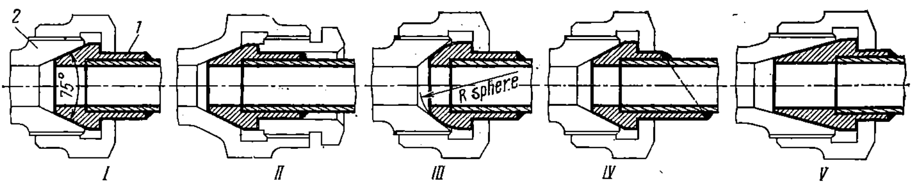


Fig. 398. Flareless-tube fittings

Ferrule *1* is soldered to the end of tubing and tightened against the conical recess in connector body *2* with a union nut (Fig. 398, *I*) or with a sleeve nut (Fig. 398, *II*). The use of union nuts makes for smaller axial and larger radial dimensions of the fitting. In contrast, sleeve nuts give larger axial and smaller radial dimensions. The included angle of cone is made equal to 75°. For better sealing, the ferrule is commonly produced from a ductile material (copper or brass).

To increase pressure in the contact zone and provide for self-alignment within certain limits, the sealing surface of the ferrule is made spherical (Fig. 398, *III*). The strength of the soldered seam is improved by slanting the end of the ferrule shank (Fig. 398, *IV*). In high-pressure applications, ferrules with an included cone angle of 30 to 40° are employed (Fig. 398, *V*).

To compensate for inaccuracies of manufacture and thermal deformations, it is advisable to bend tubing slightly; long conduits should be provided with loops or two-three coil spirals.

In flared-tube fittings (Fig. 399), sleeve *1* fitted over the tube needs no soldering to the latter. The flared end of the tube is tightened

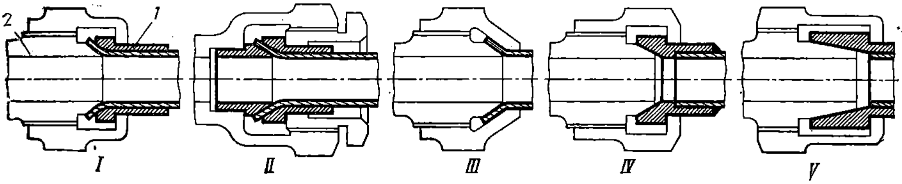


Fig. 399. Flared-tube fittings

against the cone end in connector *2* with a union nut (Fig. 399, *I*, *II*). The sleeve is normally made from steel. In a simplified design using no sleeve (Fig. 399, *III*), the flared end of the tube is pressed against the connector cone by the inside conical portion of the nut. As sealing takes place on the tube surface, the tube should be made of a ductile material.

More reliable sealing is offered by ductile-metal sleeves soldered to tube and tightened directly against the connector cone (Fig. 399, *IV*, *V*).

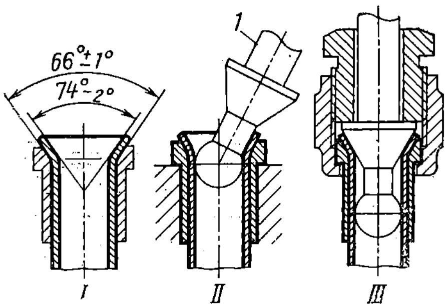


Fig. 400. Flaring a tube end

Fig. 400, *I* shows the angular dimensions of flared tube ends; such ends are made by flaring with conical plunger-type tools on presses. In short-run production or repair, the flaring is done

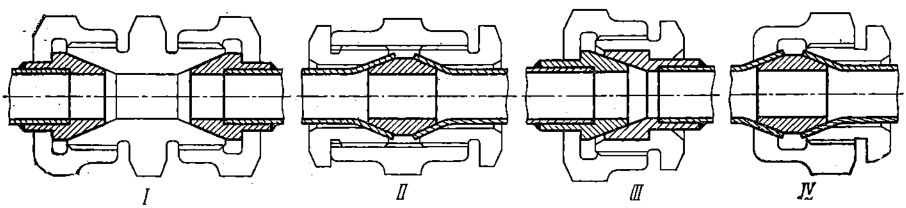


Fig. 401. Direct connection of tubes through union-type fittings

with a sphero-conical mandrel (Fig. 400, *II*). is inserted into the tube, over which the nipple is set, and revolved about the spherical-end centre; as a result, the required flare shape is obtained, which is then sized by tightening the tube end against the mandrel cone with a union nut.

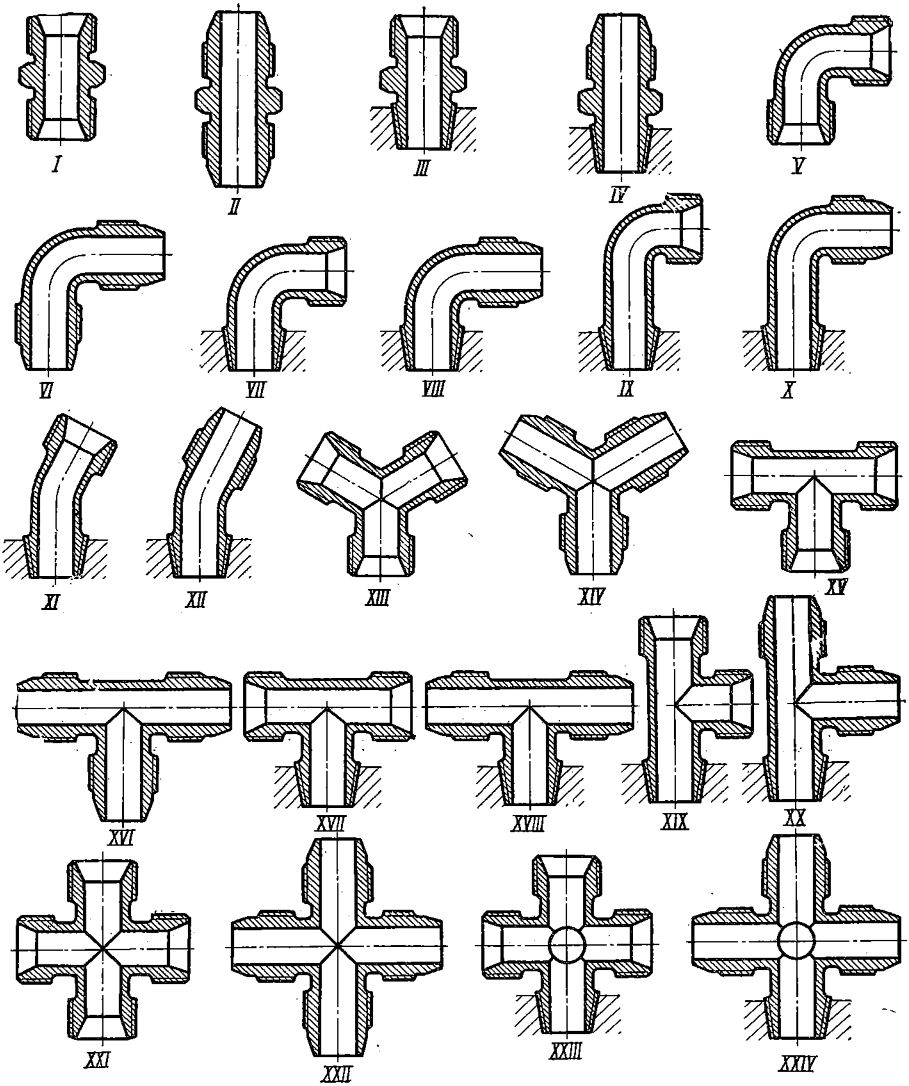


Fig. 402. Tube fittings

I-IV—nipples; *V-XII*—elbows; *XIII, XIV*—crotches; *XV-XX*—tees; *XXI, XXII*—crosses; *XXIII, XXIV*—multi-connection fittings

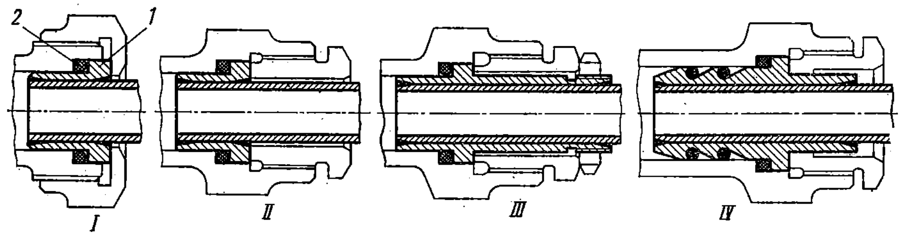


Fig. 403. High-pressure union-type fittings

Some methods of direct connection of two lengths of tubing with union-type fittings are shown in Fig. 401, *I-IV*.

Branch conduits are interconnected with nipples, elbows, tees, crosses, and other types of fittings (Fig. 402).

High-Pressure Tube Fittings. Conduits operating at high pressures are made of seamless steel tubing with a wall of 1-2 mm thick and connected through union-type fittings having flat contact surfaces (Fig. 403, *I-III*). Sleeve *1* is brazed to the tube (mainly with bronze alloys). Packing is done with *O*-rings made from elastomers or soft metals (lead, annealed copper, etc.) located in the confined space between the sealing surfaces. The joints are tightened with union nuts.

In the tube fitting illustrated in Fig. 403, *IV*, elastomeric packing rings, put into *V*-section grooves of the sleeve, operate substantially as lip packings.

Acted upon by the fluid pressure in the conduit, the rings go up the tapered portions of the grooves and become pressed against the walls of the sleeve and the body with a force proportional to the pressure.

5.4. Gland-Type Fittings

Connection of tubing with gland-type units (Fig. 404) requires no flared tube ends and can be used for tubes made of any material. The packing element, made of an elastomer, is crowded into the cavity

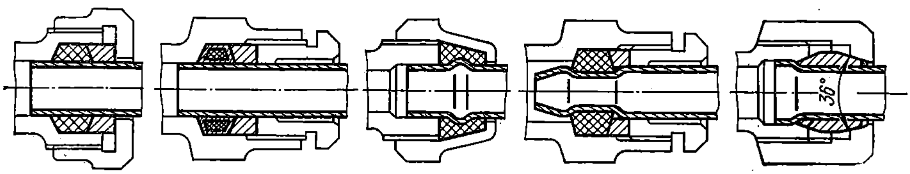


Fig. 404. Gland-type fittings

between the fitting and the tube and tightened with a union nut through a gland follower (Fig. 404, *I*). For easier assembly, the packing material is encased in a shell of soft metal (lead or copper) as is shown in Fig. 404, *II*. The fittings using plain tube ends (Fig. 404, *I*, *II*) are only applicable if tubing is fixed axially. Otherwise, it is necessary to provide the tube end with an annular crease (Fig. 404, *III*) or a neck (Fig. 404, *IV*) into which the packing element is let.

The joint using a double-cone ferrule (Fig. 404, *V*) with a rather small included angle (about 36°) is similar in principle to the described joints. The ferrule is mounted on the tube whose end is made expanded. In tightening the union nut, the ferrule squeezes the tube end, thus packing and fixing the tube axially.

5.5. Swivel Fittings

Elbow- and tee-type swivel fittings (Fig. 405) allow tubing to be disposed at any angle to the fitting in a plane perpendicular to its axis.

Swivel body *1* (Fig. 405, *I*) made in the shape of a truncated hollow sphere is tightened against a housing by means of hollow central cap screw *2*, which has radial passages connecting the cavity of the

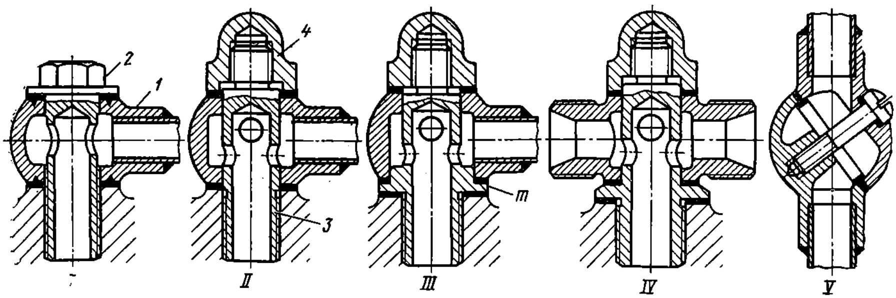


Fig. 405. Swivel fittings

swivel body with the inside of the housing. The joint faces are packed with gaskets.

With light-alloy housings, the swivel body is fitted over hollow stud connector (Fig. 405, *II*) and tightened to the housing with cap nut *4*. For suitable machining, the body cavity is made open on one side (Fig. 405, *III*) and packed on shoulder *m* of the connector. Swivel fittings are used for interconnection of several pipes (practically, up to four) or two pipes directly (Fig. 405, *IV* and *V*, respectively).

In swivel-elbow joints using tapered stud connectors, packing is achieved by tightening the body against the taper of the connector (Fig. 406, *I*, *II*). The taper is usually made with an included angle of 10 to 12°. The body and connector tapers are lapped together. To prevent the joint from overtightening, torque-limiting wrenches are used.

With this type of fitting, the tubes connected can be arranged at any angle to each other (Fig. 406, *III*).

Ball-and-socket joints (Fig. 407, *I*) provide for some angular adjustment of mutual position of the tubes being connected; the range of the adjustment depends on the amount of clearance *m* between the ball sleeve and the nut. For a reduced axial dimension, the sleeve in the joint shown in Fig. 407, *II* is made with two spherical surfaces—inner and outer.

The spherical elbow joint illustrated in Fig. 407, *III* allows for swivelling the tube into any position in a plane perpendicular to the

connector axis as well as for angular adjustment within the range determined by clearance *n*.

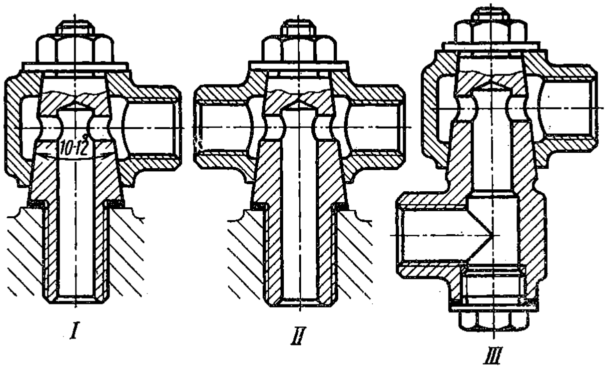


Fig. 406. Swivel fittings with tapered stud connector

Fig. 407, *IV* presents an angular connection of two lengths of tube, which can be adjusted in vertical and horizontal planes to make

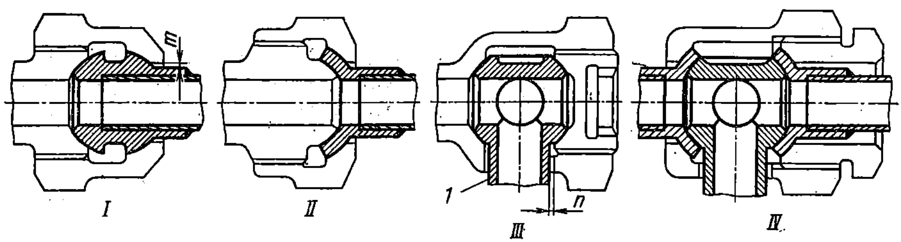


Fig. 407. Ball-and-socket fittings

a desired angle with each other and with the connector axis. The joint allows several tubes to be interconnected.

5.6. Connecting Tubes to Housing-Type Parts

Connectors are coupled to housings by soldering or welding (Fig. 408, *I*), with screw thread (Fig. 408, *II-IV*), or with flanges (Fig. 408, *V*, *VI*).

Some methods of direct connection of tubes to housings are illustrated in Fig. 409. Screwing the tube in by means of a threaded ferrule welded to it (Fig. 409, *I*) is applicable where the tube can be rotated in assembly. Otherwise, union-type and gland-type fittings are used (Fig. 409, *II-IV* and *V*, respectively). In high-pressure ap-

plications, use is made of union-type fittings with an elastic gasket compressed in a confined annular space (Fig. 409, *VI*).

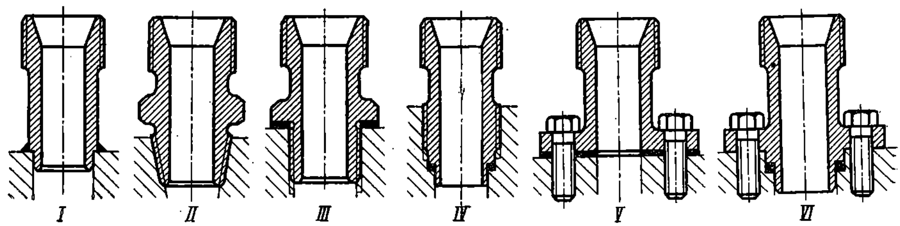


Fig. 408. Coupling connectors to housing-type parts

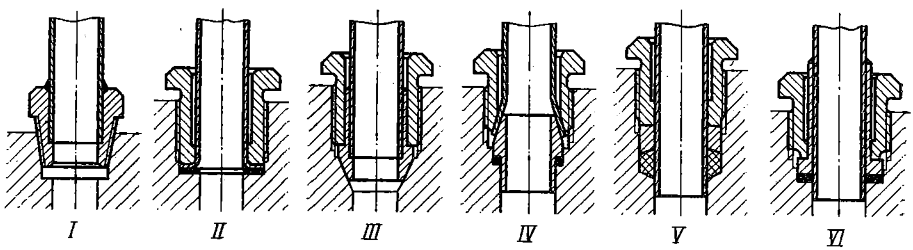


Fig. 409. Coupling tubes to housing-type parts

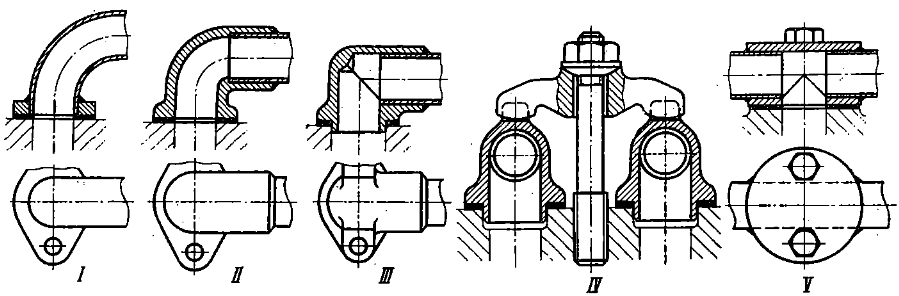


Fig. 410. Angular connections of tubing

Angular connection is effected by means of bends (Fig. 410, *I*, *II*) or elbows (Fig. 410, *III*) fastened to the housing through a flange. Elbows in twin conduits are secured with a clamp (Fig. 410, *IV*).
Fig. 410, *V* shows a joint wherein two lengths of tube are soldered into a plate, which is fastened to the housing with cap screws.

5.7. Fixing Screwed Fittings

Tightness of screwed tube fittings is often deteriorated because their components loosen with time. Therefore, all screwed fittings should be firmly secured, including those with taper thread and those tightened against elastic gaskets, which prevent loosening to some extent.

Since screwed fittings comprise essentially hollow components (nipples, union nuts, etc.) and are seldom dismantled, wiring through

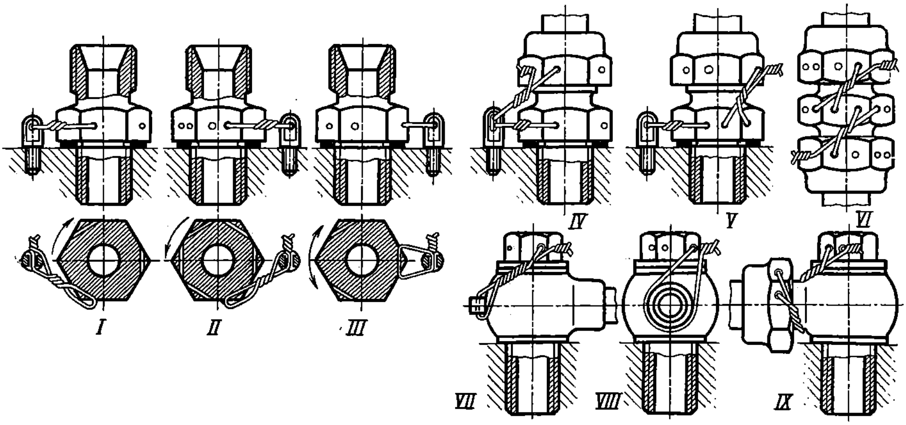


Fig. 411. Screwed fittings fixed against loosening

holes drilled near the edges of hexagons (Fig. 411) is the method most widely employed for the fixing. The wire is secured to adjacent stationary components or to special eye screws. The wire ends are interlaced with the aid of flat-nosed pliers. Three holes (Fig. 411, *I*) and, in complex units, six holes (Fig. 411, *II*) are commonly employed.

The tension arising from twisting the wire ends must induce a torsional moment directed in accordance with the hand of screwing in the component being fixed (Fig. 411, *I*). A reversely directed moment (Fig. 411, *II*) will loosen the fitting and so will the tensioning in the direction perpendicular to that of rotation of the fitting (Fig. 411, *III*). The same rule being observed, union nuts are fixed by wiring to an eye screw (Fig. 411, *IV*) or to the hexagonal portion of the screwed-in connector (Fig. 411, *V*, *VI*).

The cap-screw connectors in swivel joints are fixed by wiring to an eye made on the swivel body (Fig. 411, *VII*) or around the sleeve portion of the latter (Fig. 411, *VIII*). Fig. 411, *IX* shows a method of securing the connector and the nut together with wire turned around the swivel-body shank.

5.8. Connecting Inner Passages Together

Inner oil passages in split housings are connected through tubes whose ends carry packing rings made from oil-resistant rubber or synthetic materials (Fig. 412, *I*, *II*). To compensate for misalignment of the passages to be connected, the tube ends are made spherical (Fig. 412, *III*).

Installation of such tubes in enclosed housings is rather difficult because the free end of a tube previously mounted within one port in the housing needs to be inserted by feel into the opposite port.

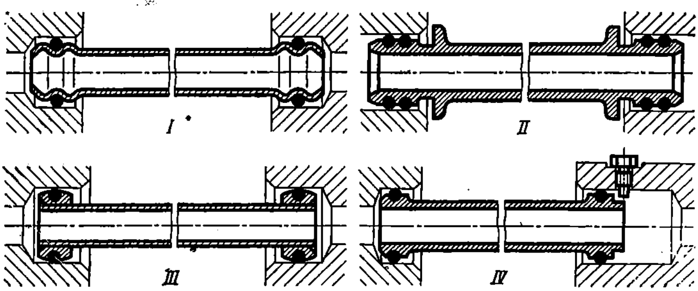


Fig. 412. Connecting inner passages

If the tube can be mounted through an opening in a housing wall, one of the ports is made sufficiently deep to allow the tube to be fit in place in the assembled housing; thereafter, the tube is fixed with a lock screw (Fig. 412, *IV*).

5.9. Hose

Flexible tube, or hose, is convenient to use for fluid conduits of complex configuration and large length. It can be disposed in hard-to-reach places outside and inside equipment and bent to fit wall configurations. Hose is fixed to a wall with tube clips or other suitable means, which should be arranged sufficiently close to each other to prevent the hose from vibration in service.

The simplest type of hose is represented by plain hose (Fig. 413, *I*) produced from plastics of adequate elasticity (polyvinylchlorides, polyamides, or polyolefines). Thick-walled hose with a wall thickness of 5 to 8 mm and internal diameter of 8 to 12 mm withstands pressures from 6 to 10 kgf/cm^2. For higher-pressure applications, use is made of hose reinforced with several layers of cord fabric (Fig. 413, *II*) or with braided tubular metal jacket (Fig. 413, *III*) which can be coated with a plastic cover (Fig. 413, *IV*).

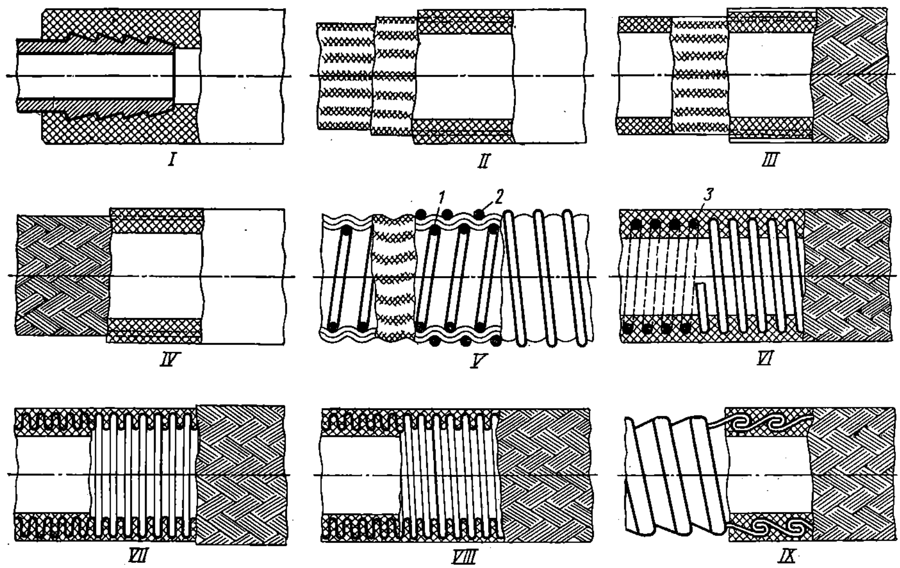


Fig. 413. Types of hose

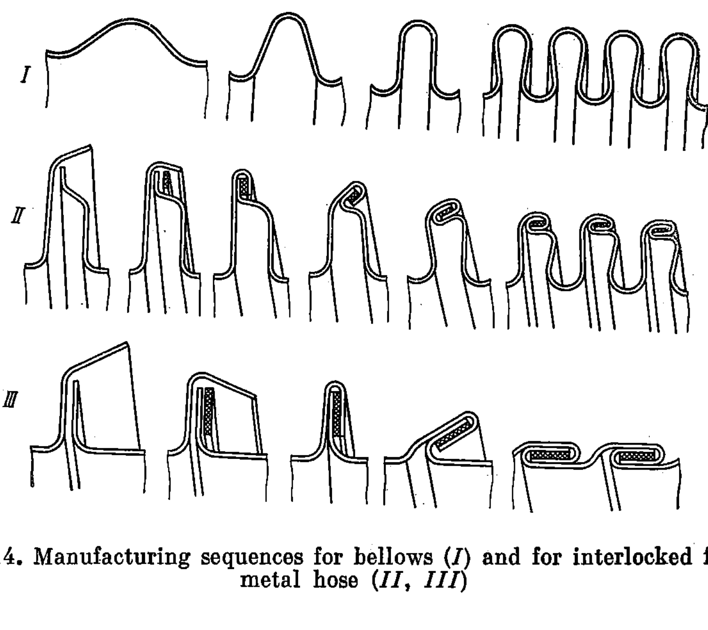


Fig. 414. Manufacturing sequences for bellows (*I*) and for interlocked flexible metal hose (*II*, *III*)

The hose illustrated in Fig. 413, *V* is produced by wrapping several layers of fabric impregnated with synthetic resin or carbinol glue around spiral carcass *1* made from zinc- or cadmium-plated wire. After wrapping, outer spiral *2* is coiled over, and the hose is subjected to hardening. In the hose shown in Fig. 413, *VI* wire spiral *3* coated with a plastic is used as a carcass.

Hose with the carcass shaped as a bellows with annular (Fig. 413, *VII*) or helical (Fig. 413, *VIII*) convolutions is the strongest. The bellows is produced from thin-walled brass or tombac tube and, for

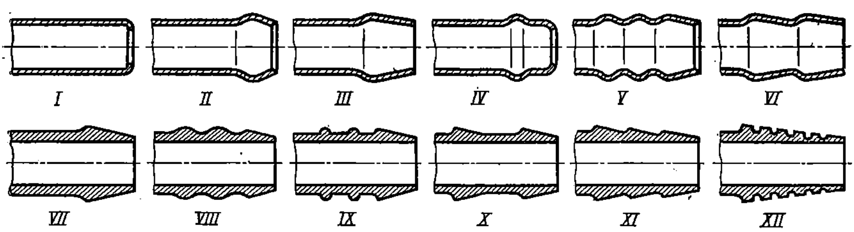


Fig. 415. Plain-hose nipple forms

use in critical joints, from Monel. In the design shown in Fig. 413, *IX*, the carcass is made in the shape of an interlocked flexible metal hose with elastic-strip packings.

Fig. 414, *I*, *II*, *III* illustrates manufacturing sequences for a bellows and for coiled interlocked flexible metal hose of two types.

The bellows and hose are commonly coated with a plastic cover and, for higher strength and wear resistance, encased in metal-wire braided jacket. This type of hose stands up to pressures of 100 to 200 kgf/cm^2.

A plain hose is connected to a housing or to another length of hose through nipples provided with annular holding crimps (Fig. 415, *I-VI*) or ribs (Fig. 415, *VII-XII*). In vacuum or low-pressure applications, the elasticity of the hose proper is sufficient for adequate anchorage. When used at higher pressures, a hose is additionally tightened on the nipple with wire or clamping collars.

Hose with a metal carcass is rigidly fixed in appropriate fittings (nipples with outer or inner tightening cones, swivel fittings, etc.). Some fixing methods for flexible hose are exemplified in Table 4.

Union nuts can be secured on thick walled nipples by the methods shown in Fig. 416. A common method, wherein the nipple is held at its shoulder (Fig. 416, *I*), is applicable if the nut can be set from the side opposite to the nipple end. Nuts fitted from the nipple-end side are fixed with snap rings (Fig. 416, *II*).

In the design shown in Fig. 416, *III*, the nut is fixed in both directions by means of snap ring *1* let into a suitable annular groove

Table 4

Hose-to-Nipple Joints

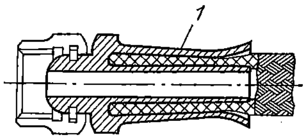
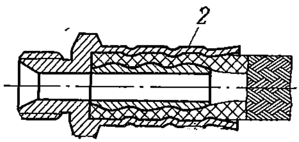
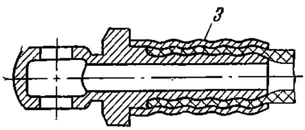
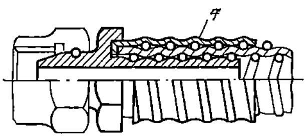
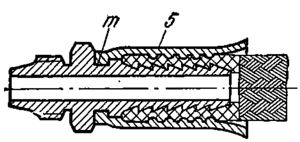
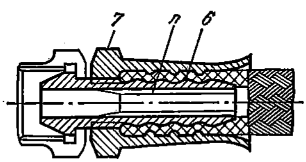
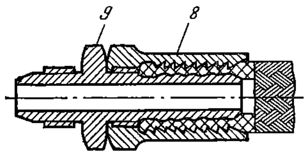
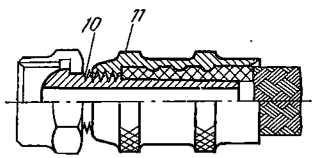
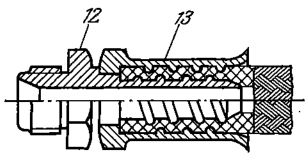
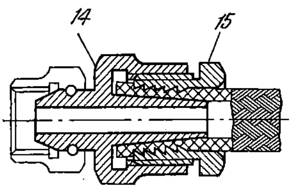
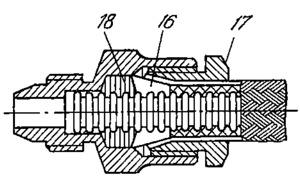
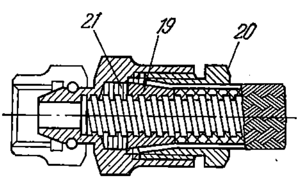
Sketch	Description of joint
	Nipple sleeve *1* is squeezed around hose end.
	Sleeve *2* is fluted to press hose around nipple shank.
	Sleeve *3* welded to nipple body is fluted to press hose around nipple shank.
	Hose with spiral wire carcass. Sleeve *4* is fluted helically to set hose tightly over helical nipple surface.
	Hose end is secured by squeezing sleeve *5* whose end portion is forced into circular groove *m* on the nipple.
	Hole of nipple *6*, screwed into sleeve *7*, is expanded by amount *n* to hold hose end.

Table 4 (continued)

Sketch	Description of joint
	Separable joint. Hose is screwed into buttress thread in sleeve *8* and then, onto tapered shank of nipple *9*.
	Separable joint. Tapered nipple *10* is screwed into sleeve *11*.
	Separable joint. Nipple *12* with a coarse-pitch thread is screwed into hose previously put into sleeve *13*.
	Separable joint. Hose is fitted over tapered shank *14* of nipple and tightened with nut *15* having buttress thread.
	Hose with bellows-type carcass. Flared end of braided jacket is set over half-ring *16* fitted onto convolutions of bellows and tightened with sleeve nut *17*. Groove packing *18* is used.
	Hose with helical-bellows carcass, which is screwed into nut *19* and tightened with sleeve nut *20*. Groove packing *21* is used.

in the nut and snapping into the groove on the nipple as the nut is fitted in place.

The nut can also be fixed with wire *2* (Fig. 416, *IV*) introduced through a hole in the hexagon into annular grooves made in the

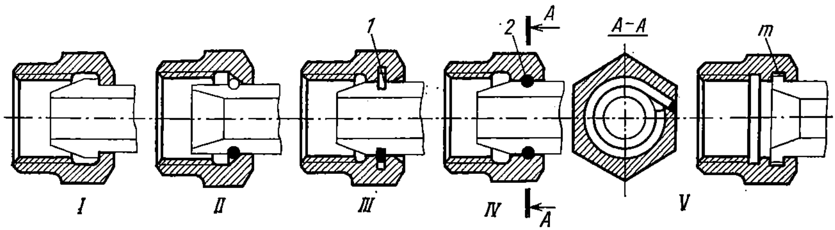


Fig. 416. Locating union nuts on nipples

nipple and the nut. The wire end that extends outward is fixed. In the joint illustrated in Fig. 416, *V*, the nut is held in place by expanding the material of the nipple end into annular groove *m* in the nut.

5.10. Quick-Release Hose Couplings

Hose conduits make use of quick-release couplings that allow two lengths of hose to be connected and disconnected by hand, without any tools.

A bayonet-locked coupling (Fig. 417, *I*) includes connector *1*, wherein pins *2* are secured, and body *3* provided with L-shaped slots

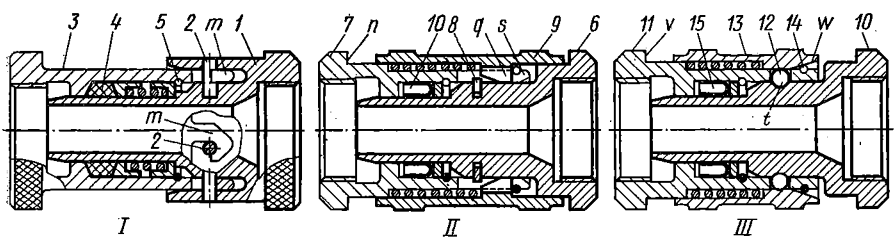


Fig. 417. Quick-release hose couplings

m into which pins *2* fit. The coupling is packed with spring-loaded packing ring *4*. To put the coupling together, the connector with the hose joined to it is inserted into the body so that the pins enter recesses between the slots and is then turned clockwise until the pins run up against the bottom of the slots; upon that, the spring will close the lock, bringing the pins into place.

For disconnection of the coupling, the connector is pushed forward, overcoming spring pressure, turned counter-clockwise so as to bring the pins out of the slots, and removed from the body. The packing ring, held back by retaining ring *5*, remains within the body.

In the coupling illustrated in Fig. 417, *II*, connector *6*, upon introduction into body *7*, is fixed therein by spring lock *8* snapping into a suitable annular groove in the body. To disconnect the coupling, spring-loaded sleeve *9* is pulled to the left up to the stop at shoulder *n* on the body; as this takes place, dogs *q*, sliding in slits *s* cut in the connector, sink spring lock *8* in its groove, and the nipple is freely removed from the body. The coupling is packed with lip packing *10*.

In the quick-release ball-locked coupling presented in Fig. 417, *III*, connector *10* is fixed in the body *11* by balls *12* located in radial openings in the body and entering annular groove *t* in the connector. The balls are locked in the groove by means of movable sleeve *13*, constantly spring-pressed into the closing position. The travel of the sleeve is restricted by stop ring *14*. The sealing means is lip packing *15*.

To disengage the coupling, sleeve *13* is withdrawn to the left up to the stop at shoulder *v* on the body, after which the connector is removed from the body, the balls being pushed out of the groove in the connector towards tapered recess *w* in the sleeve.

The balls are prevented from falling out of their seats in the body (when the connector is removed) by upsetting the material near the openings on the inner side of the body with a centre punch.

To assemble the coupling, sleeve *13* is shifted to the left and, after inserting the connector, is released and moved by the spring to the right. Running up against the balls with its inner tapered surface, the sleeve locks the coupling.

5.11. Self-Closing Couplings

In fluid conduits, the leakage of fluid during tube disconnections as well as the entry of air into the system should be prevented without the use of check valves.

Fig. 418, *I* shows a quick-release coupling that provides for losing one of the conduits connected. Spring-loaded closing disc *2* is mounted in the cavity of body *1*. While the coupling is assembled, the disc is deflected by the end of connector *3*, which abuts against the disc shank *m*. On disconnection of the coupling (the disconnecting mechanism is the same as is shown in Fig. 417, *III*), the disc is spring-pressed against seat *4*, thus closing the left-hand conduit.

The single-valve coupling presented in Fig. 418, *II* makes use of

a valve whose closing component *5* has a tapered disc portion and a cylindrical portion with openings *n* for the passage of fluid.

In a double-valve quick-release coupling designed for self-closing both conduits (Fig. 418, *III*), closing discs *6* and *7* are open in the assembled coupling because their shanks *s* and *t* abut against each other. The valve springs bear up against plates *8* and *9* having openings *v* for the passage of fluid. As the coupling is dismantled, the discs come to rest on their seats, thus cutting off the fluid flow.

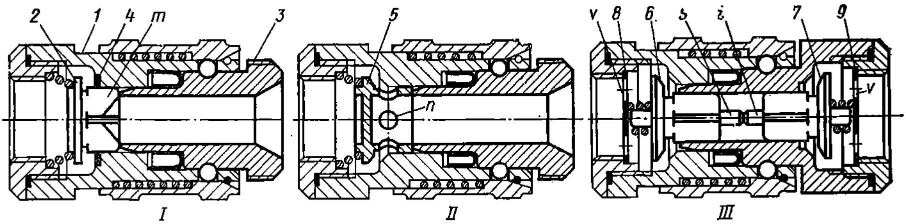


Fig. 418. Self-closing couplings

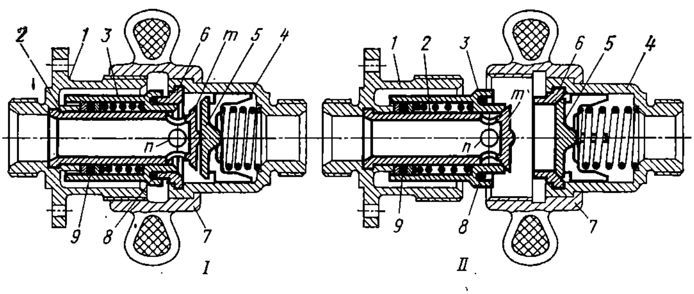


Fig. 419. Double-ended self-closing coupling

The self-closing coupling presented in Fig. 419 has valve member *2* with a conical closing disc *m* rigidly secured in body *1*. When the coupling is disconnected (Fig. 419, *II*), the valve is closed by movable seat *3* spring-pressed against the closing disc. The second body *4* holds movable disc *5* urged by its spring against conical seat *6* rigidly secured in the body.

As both bodies are put together by screwing ring nut *7* on body *1* (Fig. 419, *I*), disc *5* runs up against closing disc *m* of stationary member *2* and moves off seat *6*, overcoming spring pressure. Station-

ary seat *6* displaces movable seat *3* in the first body, which opens passages *n* in valve member *2*, whereby the chambers of both bodies are brought into connection. Packing is effected with elastic gasket *8*, placed in an annular groove in movable seat *3*, and with packing ring *9*. Both packing elements are acted upon by the spring that loads movable seat *3*.

When the coupling is being disconnected, seat *6* releases movable seat *3*, and the latter, spring-pressed against disc *m* of stationary member *2*, closes passages *n* and, thereby, the first body. At the same time, disc *5* comes out of contact with disc *m* and returns to seat *6*, closing the second body.

Chapter 6

Plastic Parts

6.1. General Data

Plastics are materials based on natural or synthetic polymers, which, when heated and pressed, are capable of taking the required shape and retaining it after hardening. Plastics are divided into *thermosetting* and *thermoplastic* types.

In addition to polymers, plastics may contain mineral or organic fillers, plasticizers, stabilizers, pigments, lubricants, etc.

Plastics feature low density, high electric and heat-insulating properties, stability to atmospheric effects, resistance to aggressive media and sharp temperature changes.

Heat resistance of plastics is not high. For most plastics, the Martens temperature amounts to 80-140°C. Some plastics (e.g. polysiloxane) have a Martens temperature of 200 to 250°C.

The Martens temperature is defined as the temperature at which a standard specimen subjected to loading with a specified bending force develops residual strain. Hence, the Martens temperature characterizes the stability of shape at elevated temperatures under load.

In many cases, the maximum permissible working temperature is determined by other factors than the extent of distortion of the material. These factors depend on operating conditions; an instance is deterioration of dielectric properties with increased temperature. For parts operating in no-load or small-load conditions, the maximum permissible working temperature can be the temperature whose prolonged action upon the material causes its thermal degradation. This temperature may be much higher than the Martens temperature.

With thermosetting plastics, heat resistance is determined from the mass lost by a specimen which is held at a specified temperature over a long period of time.

Serious drawbacks of plastics as structural materials are low hardness (on the overage BHN 10-30) and low mechanical strength.

With most plastics, the tensile breaking strength amounts to 5-10 kgf/mm^2. The introduction of fillers in the form of fibres, fabrics, and laminates increases the strength to 20-30 kgf/mm^2. The strongest plastics are glass-fibre reinforced laminates, whose tensile breaking strength, being 40-50 kgf/mm^2, is comparable with that of carbon steels.

Another drawback to plastics is their low modulus of elasticity and hence low stiffness of plastic parts. With most plastics, the elastic modulus amounts to $E = 100\text{-}300$ kgf/mm^2; fillers increase it to $E = 700\text{-}1{,}000$ kgf/mm^2. Glass filled plastics have $E = 1{,}500\text{-}3{,}000$ kgf/mm^2, yet it is lower by a factor of 8 to 15 than the elastic modulus of steel.

Impact strength of plastics is also low. Most plastics have a specific impact strength a_H of 0.1-0.3 kgf·m/cm^2, and only few of them, such as tetrafluoroethylene, polycarbonate, capron and also glass-fibre reinforced materials have $a_H = 1\text{-}3$ kgf·m/cm^2.

Nearly all plastics have low stability of shape due to low stiffness and relatively high softness (resulting in shape changes under load), high linear expansion (resulting in dimensional variations due to temperature changes) and rapid softening at elevated temperatures (with thermoplastics). Many plastics swell in water, kerosene, petrol, and mineral oils. Some plastics (e.g. polytetrafluoroethylene) creep under relatively low stresses (0.2-0.5 kgf/mm^2) even at modest temperatures (20-60°C).

Mechanical strength of plastic products is greatly affected by moulding conditions, so that variations in a moulding process cause variations in product strength within a batch of moulded parts. In complex-shape parts, strength variation is due to a nonuniform structure resulting from different conditions of moulding and solidification of the material in different sections of the part.

When subjected to a prolonged action of high or low temperatures or sharp temperature changes, most plastics gradually lose their initial properties, becoming weak and brittle. Exposure to ultraviolet radiation (direct sun rays) over a long period of time also makes plastics brittle and bleaches coloured plastic materials.

Resistance to light and atmospheric effects can be improved to some extent by introducing special stabilizing agents into plastics. Some varieties of plastics (e.g. tetrafluoroethylene) are completely atmosphere resistant.

Plastics Compositions. Synthetic resins are used pure or with fillers. The latter are applied in the form of powders, fibres, fabrics, and laminates. Fillers amount to 20-60% of the product mass.

Powdered fillers are introduced into *moulding powders* used for moulding complex-shape products. Wood flour, kaolin, powdered quartz, mica, graphite, metal powders, etc. are used as fillers. The most commonly employed binders are phenol-formaldehyde resins. Moulding powders based on aminoplasts are a recent development being now in use.

Strength and toughness of plastics are improved by kaolin added as a filler; heat resistance, by asbestos, and dielectric properties, by mica and powdered quartz. Metal powders increase thermal conductivity and strength, and powdered graphite, antifriction properties.

Fibre fillers (cotton flocks, glass fibre, etc.) are used to increase tensile and impact strength. Glass-fibre reinforced plastics, which are compositions of synthetic resins with glass fibres 5 to 10 μm thick featuring great strength and high elastic modulus, have found the most extensive use among fibre filled plastics. Glass fibres increase the strength of plastics 3 to 4 times.

Fibres may be used continuous or chopped, oriented or nonoriented.

Glass-filled plastics with oriented fibres (anisotropic materials) are the strongest, with the strength along the fibres being 2 to 3 times that across the fibres.

Binders for this family of plastics include phenol-formaldehyde resins, epoxy resins, silicones, etc.

Glass-filled plastics are used to produce highly loaded shell-type structural components, such as hulls of lightweight vessels, cabins, car bodies, etc. Plastics filled with oriented glass fibres are used for high-strength plates and tubes.

Compositions of synthetic resins with asbestos fillers are used for parts which should have high heat resistance.

Maximum thermal conduction is provided by compositions of silicone resins with quartz fibres.

Cloth laminates are obtained by hot pressing of oriented sheets of cotton fabric impregnated with curing synthetic resins.

Binders for cloth laminates are usually phenol-formaldehyde resins mixed with polyvinyl-acetate, silicone and epoxy resins.

Paper laminates are compositions of synthetic resins with paper, and wood laminates those with veneer, obtained by pressing. The most common binders are phenol-formaldehyde resins. In this family of plastics wood laminates, have found the greatest use; their fields of application are construction, where they are used as facing materials, and high-loaded shell-type structures. Wood laminates have a density of 1.3-1.5 kg/dm^3, tensile strength of 20-30 kgf/mm^2, specific impact strength of 1-2 kgf·m/cm^2, elastic modulus of 1,500-2,000 kgf/mm^2, heat resistance 140-160°C, and water absorption of 5-10%.

Wood laminates are also used to make fan and propeller blades, vanes for the first stage in axial compressors, and bearings subjected to high loads at slow and medium shaft speeds.

Foamed plastics are expanded resins with uniformly distributed cells. The expansion of synthetic resins is effected by introducing therein *foaming agents*, which, at moulding temperature, release inert gases in large quantities. Ammonium carbonate is the most commonly used foaming agent. A uniform distribution of cells is provided by adding emulsifiers. The cells take from 80 to 98% of the material volume. The porosity and cell size depend on the quantity of foaming and emulsifying agents, on the properties of the resins, and on moulding conditions.

Depending on the basic resins, the foamed plastics can be stiff or elastic. The first group is obtained from thermosetting resins (phenol-formaldehydes, aminoplasts) and cold-set polymers. The second group is obtained from elastic thermoplastics (polyvinylclorides, polyolefins). Elastic properties of foamed plastics can be changed by combining various resins.

Foamed plastics feature low density (0.02-0.3 kg/dm^3), very low thermal conductivity (0.03-0.06 cal/m·h °C) and good heat and sound insulation properties. The strength of foamed plastics, being lower than that of solid plastics, is inversely proportional to the porosity.

Two varieties of foamed plastics are distinguished: with closed gas-filled cells and with open interconnected cells. The first are known as *closed-cell foamed plastics*, and the second, *open-cell foamed plastics*.

Closed-cell foamed plastics have greater strength, better sound and heat insulating properties, and stronger resistance to the action of different media than open-cell foamed plastics.

The latter, when obtained from elastic polymers, are mainly used as shock-absorbing materials, for making soft seats, for elastic mountings of instruments, etc.

Closed-cell foamed materials are generally used in construction for heat and sound insulation purposes. They are obtained from polystyrene and polyvinylchlorides in the form of plates and blocks. Their heat resistance is within a range of 60-80°C.

Closed-cell foamed plastics are also used to fill shell-type structural elements for greater strength and stiffness. These materials have found wide application in the aircraft industry for filling recesses in aircraft sections, fairings, elements of tail assemblies, helicopter rotors, seaplane floats, etc. By providing connection between the walls of structural elements, the foamed plastics filling makes for uniform distribution of operational loads among these elements, markedly increases their stiffness and stability, and makes it possible to reduce the number of internal metallic links (ribs and stringers) or dispense with them completely.

In building shell-type structural elements, use is usually made of self-blowing plastics in the form of semiliquid mixtures of resins with foaming, emulsifying, and hardening agents. Such a mixture is poured into the space between the structure walls and then heated. As a result, the plastic is foamed and hardened. Self-foaming materials for shell-type structures must have good adhesion to the base metal, high strength and stiffness. Since the strength of foamed plastics depends on porosity, materials of a 80-90% porosity (or a density of 0.2-0.3 kgf/dm^3) are used.

Self-blowing plastics are obtained mainly from phenol-formaldehyde compositions. Use is also made of polyurethanes and silicones.

Foamed plastics based on silicones can withstand temperatures up to 150°C.

Strong, stiff and lightweight structures are also obtained with the use of honeycomb-type plastics, which are produced by joining cotton or glass-fibre fabrics preformed as honeycombs and impregnated with thermosetting or thermoplastic resins. The honeycombs measure from 8 to 12 mm.

The maximum strength and stiffness are typical of metallic honeycombs, obtained by adhesive-bonding preformed foils of aluminium or magnesium alloys, preliminarily coated with a film of phenolneoprene adhesives or modified epoxy adhesives. The same adhesives are used for bonding outer skins to honeycomb layers.

Machining of most plastics is not easy since their innate wear resistance causes rapid wear of the cutting tool. Plastics are machined by carbide or diamond tools at high speeds and small feed rates. Machined plastic parts are inferior in strength and surface finish to compression and injection moulded parts.

Plastic parts are predominantly articles of mass production, where the use of costly moulds, presses, and moulding machines is justified. Single-piece production of such parts is slow and inefficient. An exception is the manufacture of large shell-type structures from glass-filled plastics. This process hardly lends itself to mechanization and has to be carried out manually.

The merit of plastics is that they can be easily reinforced with metallic inserts in the process of moulding. This makes it possible to develop parts composed of metals and plastics.

In the initial state many plastics (aminoplasts, polystyrenes, polyvinylchlorides, polymethylmethacrylates) are transparent or white and can be readily coloured.

Application of Plastics. Plastics are very important as structural materials in modern technology. Their main fields of application are electrical and radio engineering, chemical machinery and instrument making.

Wear-resistant plastics of the polyamide and polyolefin types are used for straight slideways in machine tools. When protected against abrading substances (swarf, dust, dirt, etc.), plastic slideways are capable of functioning over a long period of time even with inadequate oiling.

Low mechanical strength and rigidity and inferior stability of shape are the factors that disqualify plastics for high-load applications. Only glass-filled plastics can be used for loaded shell-type structural elements.

Plastics are used to make separators of antifriction bearings. The separators are either compression moulded or injection moulded. The mouldings are finished by removing flash, the methods of removal being other than mechanical (since burrs left after machining may

damage the bearing during service). The most acceptable method is treatment with flame. Separators must be dimensionally stable, which is achieved by special methods (boiling in oil, etc.).

Formerly, the separators were made of cloth laminates only. At present, use is made generally of teflon (polytetrafluoroethylene), sometimes of expanded teflon, which after saturation with oil becomes self-lubricating. Bearing separators having a thin antifriction coating of plastics have found wide application. The coating should be no more than 0.3 mm thick. To reduce friction, plastics used for separators are filled with graphite or molybdenum disulphide.

6.2. Methods of Making Plastic Parts

Sheets and plates of plastics are mainly obtained by *calendering*, or pressing between rollers on multi-roller machines. Calendering is also used to produce corrugated sheets for honeycombs. Form parts are obtained from sheets by pressing in moulds with a rigid or elastic punch (an inflated rubber bag).

Extensive use is made of pneumatic and vacuum forming. According to the first method, a sheet heated to a plastic condition is clamped along the mould perimeter, and then forced into the mould cavity by compressed air. In accordance with the second method, vacuum is created in the mould so that the blank is drawn therein and pressed against the surface of its cavity. Thin-walled parts such as covers, casings, containers, fairings, etc. are produced in this way.

6.2.1. Compression Moulding

Compression moulding is used to produce form parts from thermosets and thermoplastics. The usual form of basic material is pellets, granules, etc., and, for products with powdered fillers, moulding powders. The process is effected in moulds, comprising the female and male dies, at elevated temperatures (the mould is heated) to ensure the curing of the material.

A measured quantity of preheated moulding powder is charged into the female die, whereupon the male die is brought down mechanically or hydraulically to press the material. The latter is held under pressure for a short time whereby it is cured. The male die is then retracted, and the part is removed from the female die by ejectors.

The moulding conditions (preheating temperature, processing temperature and pressure, and curing time) depend on the moulding compound and on the shape and size of the part to be formed. Usually, the preheating temperature is 130-180°C, the processing temperature is 200-220°C, the moulding pressure is 100-300 kgf/cm^2, and the curing time is 15-30 s.

At present, compression moulding is done on automatic multistation rotor-type presses with automatic high-frequency heating, which are capable of producing 100 pieces per minute and over.

The moulded part depends for its dimensional accuracy on the accuracy of the mould and of the material batching, and on variations in moulding conditions.

The process provides high surface finish. Where the mould cavity is appropriately finished (chrome plated and polished), a surface finish of 0.080-0.160 μm Ra can be obtained on the product.

6.2.2. *Injection Moulding*

This process is used to form parts from thermoplastics. The basic material (in the form of granules, pellets, etc.) is heated until it is fully softened. The liquid mass obtained is fed into a heated cylinder wherefrom it is ejected by the piston through the runners into the cavities of a cooled metallic mould. After the mouldings have cooled and solidified, the mould is opened and they are ejected therefrom. The runners, gates and flash on the mouldings formed at the mould parting face are cut off and their traces are smoothed out. The softening temperature of the material being processed depends on its composition. The moulding pressure is usually 1,000-1,500 kgf/cm^2, the mould temperature is 20-40°C.

Injection moulding provides for greater output and higher quality of products than press moulding. Thus, for instance, a surface finish of 0.020-0.040 μm Ra can be achieved.

Modern injection-moulding machines with multi-cavity moulds and fully automatic working cycle are capable of making up to 200 mouldings per minute.

To eliminate internal stresses and improve the uniformity of structure, the mouldings are heat treated, i.e. heated in the absence of air (usually in mineral oil) to 140-160°C, held for 1.5 to 2 hours and then slowly cooled.

6.2.3. *Extrusion*

Extrusion is used for making bars, tubes, hoses, plates, films, special-section products (e.g. hand rails, skirting boards, etc.) out of thermoplastics. The process is carried out on screw extruders. The material is delivered through a feed bin to the heated screw cylinder where it is engaged by the screw (also heated); the latter moves the material along the cylinder, mixes and compacts it. The compacting is effected by reducing the pitch or the height of the screw thread. An extrusion die with an aperture corresponding to the product cross section is located at the exit end of the cylinder. The formed product coming out of the die as a continuous bar undergoes cooling. When solidified, it is cut to lengths.

A recent development is the use of the heat resulting from the friction of the material against the cylinder wall and the screw for heating this material. This method, known as adiabatic extrusion, simplifies extruders and increases efficiency of the process.

Extrusion is widely used to make insulation coatings on conductors, cables, etc. The conductor to be coated is fed from a coil through the central bore in the screw to the extrusion die where it is coated with the plastic material.

To produce plastic films, an angular die head is installed at the exit end of the extruder. The material comes out of the die in the form of a thin-walled tube, then it is turned at right angles, blown by compressed air to obtain walls of the specified thickness and moved into a wedge gap between two endless bands where it is flattened out. The double plastic band thereby produced is moved on by rollers to a cutting device.

Tubular blanks obtained by extrusion are used to produce (by blowing in dies) container-type articles (flasks, bottles, etc.). The bottoms are then welded.

6.2.4. Forming Glass-Filled Plastics

Small glass-filled plastic parts are obtained by hot press moulding in metal moulds. This method is impracticable for large parts, since it requires powerful presses and costly large moulds.

Large shell-type structural parts are usually produced by spraying semi-liquid plastic filled with fibre glass onto the surface of a pattern. The plastic and the chopped glass fibres are suitably mixed and fed to a sprayer. The plastic ejected from the sprayer is applied to the pattern until a layer of the specified thickness is obtained.

The patterns representing the inner contour of a product are used where a smooth and accurate internal surface must be obtained. The patterns that represent the outer contour are used to obtain a smooth and accurate external surface.

When making parts from cold-setting plastics, the patterns are made of wood, gypsum, cement, and also of thermosetting plastics. Heated metal patterns are used for plastics hardened at high temperatures. The deposited layer is compacted by rollers or by elastic heat-resistant rubber or silicoplast bags filled with compressed air. After the layer has solidified, its surface is finished, primed, and coated with a synthetic varnish.

The dimensional accuracy of plastic parts obtained by spraying is not high. Large parts may have size variations up to several millimeters. Such parts are also inferior in strength to parts produced by hot press moulding.

Hollow parts shaped as solids of revolution (tubes, cones, etc.) are produced by a method whereby continuous glass fibres impregnat-

ed with a synthetic resin are wound on a rotating mandrel. The glass-fibres feeder is mounted on a carriage reciprocating relative to the mandrel. In this way the mandrel is coated with several criss-cross layers, which are compacted with rollers.

High-strength plates with oriented glass fibres are produced by winding the latter on a large-diameter roller. The cylinder obtained is cut along its generator before it has hardened, straightened out and then pressed in flat or form dies.

6.2.5. Welding Plastics

All thermoplastics weld fairly well. High-elasticity plastics (polyolefines, polyamides, polymethyl methacrilates, etc.) are welded without additional welding material. Thin sheets and films are lap welded by passing these between electrically heated rollers. Plates, rods and the like are butt joined. Their surfaces are pressed against each other at a pressure of 1-3 kgf/cm^2 and heated by high-frequency currents or ultrasound. The strength of the weld is about that of the welded materials.

Plastics of lower elasticity (vinylplastic, teflons, etc.) are welded with the use of welding sticks of the same material as the parts to be welded but with the addition of plastisizers. The edges being joined are prepared to form a weldpool. Welding is effected by a jet of hot air. The weld's strength is 70 to 80% that of the joined materials.

Methods for welding thermosetting, cold-setting, and glass-filled plastics have also been developed.

Plastics are well bonded with adhesives which are the solutions of a given polymer in a suitable solvent. Some adhesives (e.g. polyvinylacetate, phenol-neoprene, epoxy compounds, etc.) are universal, and can be used for bonding plastics to metal, glass, ceramics, etc.

6.3. Designing Moulded Plastic Parts

Design requirements placed on plastic parts are generally similar to those placed on metal castings and stampings. The main efforts should be concentrated on product design measures aimed to simplify the manufacture of costly moulds, increase the output of moulding operations, provide for uniform properties of the material throughout the moulded part, and eliminate internal stresses therein. Whenever possible, moulded parts should be so designed as to obviate the need for subsequent machining operations (except for flash removal).

When designing a moulded plastic part, the departure point is to locate the parting line determining the part shape, to establish the direction of draft, and to arrange holes and inserts.

As a rule, the mould should have a single parting face. Several parting faces in different planes complicate its design. Additional

parting faces perpendicular to the main one are especially objectionable.

To facilitate their manufacture, moulds should be as simple in shape as possible (e.g. with cylindrical, conical, and other surfaces obtained by turning).

Complex-shape moulds are much harder to machine; they require copying milling and sometimes hand working. The forming surfaces of such moulds are finished by electrical and hydraulic polishing. It should be borne in mind that convex mould surfaces forming internal surfaces of the part are simpler to machine than cavities forming the part external surfaces. For this reason, all complex-shape features are best to arrange on the internal surfaces of the part, whereas its external surfaces should be as simple in shape as possible.

6.3.1. Manufacturing Requirements for Parts Design

The shape of a moulded part should allow for easy removal of the part from the mould. Local projections and recesses on the part sides should be avoided since they require additional parting lines, special moulds with side cores retracted in a direction perpendicular to that of the part removal, detachable mould elements, etc. These measures result in complex special moulds and prevent the use of high-output unit-built moulds. Local lateral projections and recesses are particularly undesirable (and in small parts absolutely impermissible) on the part internal surfaces.

Examples of inconsistent designs requiring complicated moulds are given in Figs. 420-423. The same figures illustrate correct designs adapted for production in moulds with a single parting face.

The internal and external surfaces of moulded parts should not have recesses and grooves which run parallel to the mould parting face. An instance of a wrong design is given in Fig. 424. The part in the position shown in Fig. 424, *I* cannot be formed at all; when a part is formed in the position shown in Fig. 424, *II*, flash in the groove on the parting line is hard to remove.

Raised surfaces should be so arranged on the part that they can be formed in one half of the mould only. Fig. 425 shows a ball grip with a knurling. When moulded as shown in Fig. 425, *I*, it has the knurling situated in both halves of the mould, the latter being hard to manufacture. First, it is difficult to ensure that the knurls in both halves of the mould coincide; in addition, a flash which is hard to remove develops on the knurling along the parting line. The correct arrangement is shown in Fig. 425, *II*.

Internal and external walls should have draft or taper (Fig. 426) providing easy removal of the part from the mould.

An exception is surfaces whose functional requirements permit no taper, e.g. the flanks of spur-gear teeth, which are usually arranged

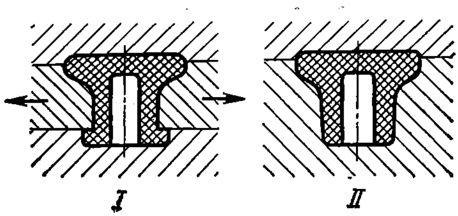


Fig. 420. Improper (*I*) and proper (*II*) shapes of control knob (shape *I* requires a mould with retractable sides)

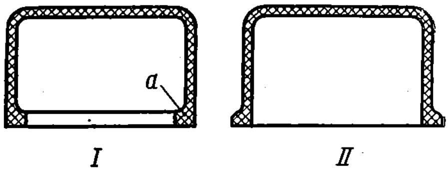


Fig. 421. Improper (*I*) and proper (*II*) shapes (bead *a* complicates the mould)

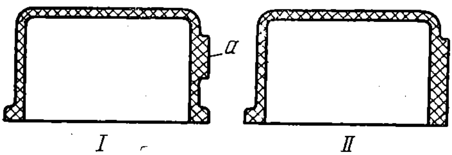


Fig. 422. Improper (*I*) and proper (*II*) shapes (boss *a* complicates the mould)

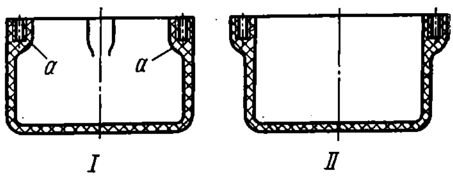


Fig. 423. Improper (*I*) and proper (*II*) shapes (bosses *a* complicate the mould)

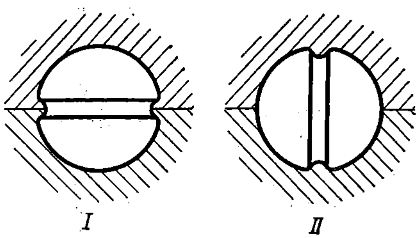


Fig. 424. Moulding parts with circular grooves

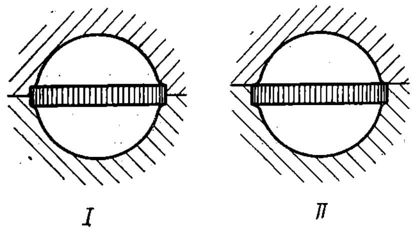


Fig. 425. Moulding parts with raised surfaces

I—incorrect; *II*—correct

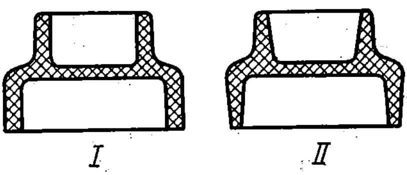


Fig. 426. Moulding drafts

I—incorrect; *II*—correct

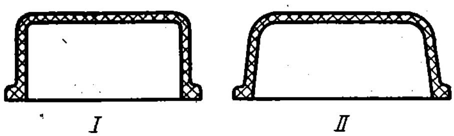


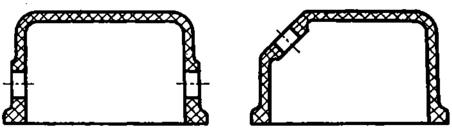
Fig. 427. Parts designed with (*II*) and without (*I*) inclined walls

parallel to the direction of the part removal. Such surfaces should be as short in this direction as possible.

Minimal amounts of taper for general purpose parts depending on wall height *h* are as follows:

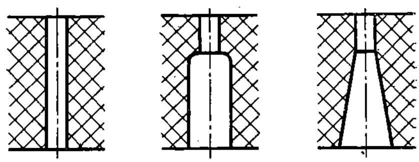
wall height, mm	10	10-50	50-100	100-200	200
taper	1 : 10	1 : 20	1 : 50	1 : 100	1 : 200

Draft on internal walls should be greater then on external walls to facilitate stripping of the part, which shrinks and tightens up on the mould forming elements. It is advisable, wherever possible, to provide parts with tapered, pyramidal and similar surfaces (Fig. 427).



I *II*

Fig. 428. Holes arranged square (*I*) or oblique (*II*) to the direction of mould opening



I *II* *III*

Fig. 429. Incorrect (*I*) and correct (*II*, *III*) hole configurations

That helps to remove the part from the mould and reduce the internal stresses developed in the walls as a result of shrinkage taking place as the material hardens.

Moulded parts should be provided with bearing surfaces for mould ejectors. Usually, such part features as bosses, lugs, etc. are used as bearing surfaces; but sometimes special surfaces 4 to 6 mm in diameter have to be developed on the part. These surfaces must be perpendicular to the direction of ejection.

In small parts, bearing surfaces for ejectors should best be located at the part centre. In large-area parts (such as covers, guards, etc.) provision should be made for several bearing surfaces arranged symmetrically along the periphery. These surfaces should be located at the regions of increased stiffness or must be reinforced with ribs in order to exclude distortions or breakage of the part during its ejection from the mould.

Holes in plastic parts are formed by cores installed in the mould.

The following rules must be observed when arranging holes in moulded parts. Holes arranged at right angles or inclined to the mould's parting face (Fig. 428) should be avoided, since otherwise the mould will be more complex in design (the cores will have to be removed prior to ejection of the part). In some cases cross holes in a moulded part are more advantageous to machine.

The length-to-diameter ratio (for both blind and through holes) should be no more than 3 to 5, the minimum permissible hole diameter should be 0.8 to 1 mm.

For greater stiffness, cores (especially long and small in diameter) are well to provide with a partial increase in diameter over the maximum possible length, reducing the length of the hole of the specified small diameter to a minimum (Fig. 429).

Bosses with holes should have sufficiently thick walls to prevent their cracking due to shrinkage (Fig. 430).

Holes are usually arranged at right angles to the end faces, without chamfers or corner rounding (Fig. 431, *II* and 432, *II*), the latter requiring complicated cores. The chamfers and corner rounding are permissible only at the hole end where the core is attached to the mould (Figs. 431, *III* and 432, *III*); when disposed at the opposite end, they make the parting of the mould components practically impossible.

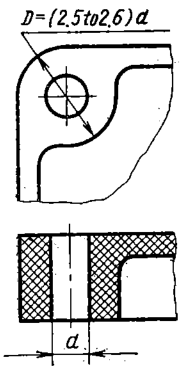


Fig. 430. Determining the thickness of walls for bosses

Threaded Holes. Forming threaded holes directly in plastic parts should be avoided. Such threads require the use of screwed-out cores, which calls for more complex moulds and more time to open the mould. In addition, threads formed in plastic parts are weak, they wear out fast and get crushed with repeated screwing and unscrewing. The depth of threaded holes should be no less than 2.5-3 thread diameters.

Threaded holes should be provided with chamfers or recesses (Fig. 433). Generally, however, threaded holes for screw fasteners are more advantageous to have in metal inserts moulded into plastic parts.

In some situations, threaded holes can be transferred from a plastic part to the mating metal part. An instance is given in Fig. 434, which shows a plastic star knob fastened to a metal rod. In the assembly of Fig. 434, *I*, thread is formed in the handle, in those of Fig. 434, *II* and *III* it is cut in the rod, whereas the knob has a plain hole. The merit of the design shown in Fig. 434, *III* is that the hole for the fastener is arranged square to the mould parting face and consequently can be formed by a permanently set core.

In plastic parts such as threaded plugs and caps the thread should have a coarse pitch and the minimum number of threads. The rounded thread profile (Fig. 435) is recommended. The threaded hole should have a chamfer or recess, the first turn (nearest to the end face) should be gradually reduced in height and smoothed out completely.

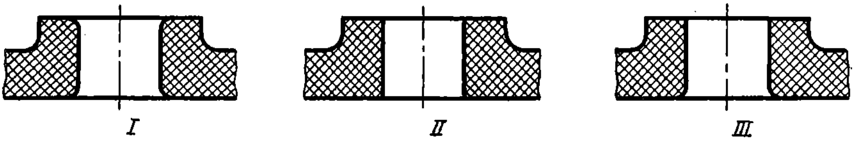


Fig. 431. Configuration of holes
I—incorrect (with corner rounding at both ends); *II*—correct (without rounding); *III*—permissible (rounding at the end where forming core is attached to mould)

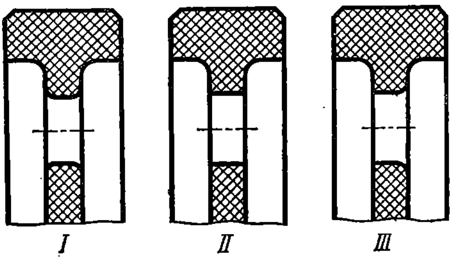


Fig. 432. Incorrect (*I*), correct (*II*), and permissible (*III*) hole configurations

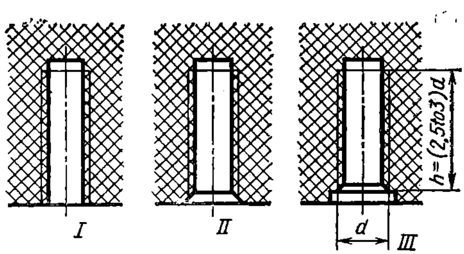


Fig. 433. Holes with moulded threads
I—incorrect (threads come to moulding face); *II*, *III*—correct (with chamfer or recess)

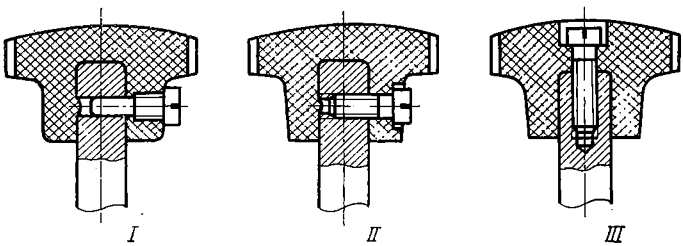


Fig. 434. Securing handle to rod
I—incorrect; *II*, *III*—correct

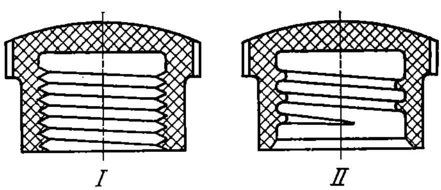


Fig. 435. Thread in moulded cap
I—incorrect; *II*—correct

Wall Thickness. A uniform structure and physico-chemical properties of the plastic material at different points of the part depend mainly on such factors as an evenly filled mould and uniformly cured moulding. The walls of a moulding should be as uniform in thickness as possible, local heavy sections should be avoided (Figs. 436-439). It has been found from practical experience, that the difference in thickness between the walls of a moulding should not exceed a 1 : 3 ratio. A smooth junction should be provided between the walls of different thickness (Fig. 440).

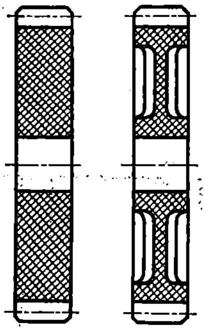


Fig. 436. Gearwheel construction
I—incorrect; *II*— correct

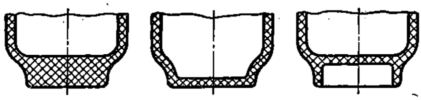


Fig. 437. Eliminating local heavy section

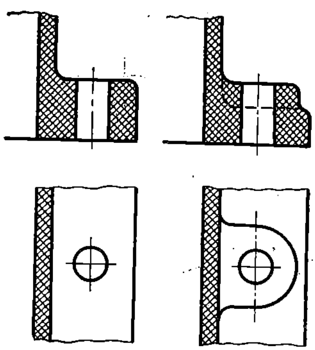


Fig. 438. Incorrect (*I*) and correct (*II*) configuration of bosses

The corners formed by mutually perpendicular or inclined walls should be rounded to a maximum possible radius (Fig. 441) for better filling of the mould.

The moulding walls should be of medium thickness. The wall thickness in excess of a certain limit causes a nonuniform structure of the material across the wall and thereby weakens the part. The mean thickness of walls for complex-shape parts can be found approximately from the formula $s = (0.25\text{-}0.5)\sqrt{L}$, where L is the maximum overall dimension of the part, mm (Fig. 442).

Ribbing. The strength and stiffness of moulded parts should be increased by rational ribbing rather than by thickening the walls. When arranging ribs, it is necessary to follow definite guidelines. The thickness of ribs should be 0.6-0.8 that of the walls, but no less than 0.8-1 mm in small parts. The height-to-thickness ratio for ribs should not exceed 3 to 5. The surface of ribs should be inclined towards the mould parting face. Corners formed

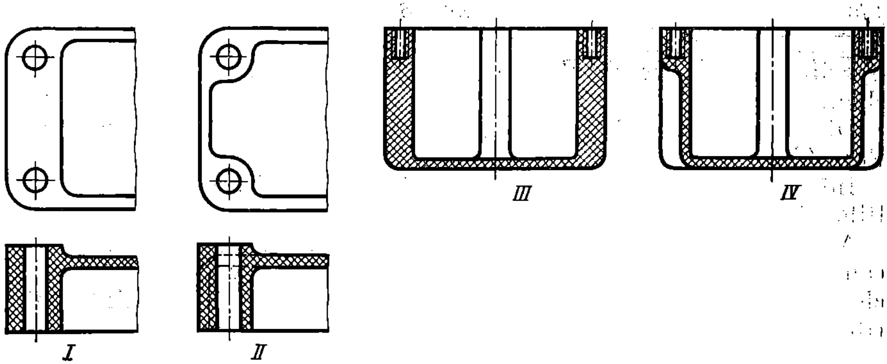


Fig. 439. Eliminating local heavy sections. Improper (*I*, *III*) and proper (*II*, *IV*) design

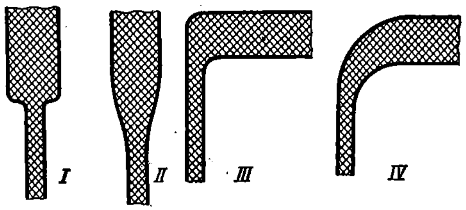


Fig. 440. Junction of walls different in thickness

I, *III*— incorrect; *II*, *IV*—correct

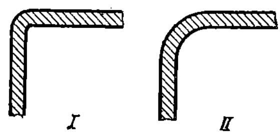


Fig. 441. Corner rounding

I—improper; *II*—proper

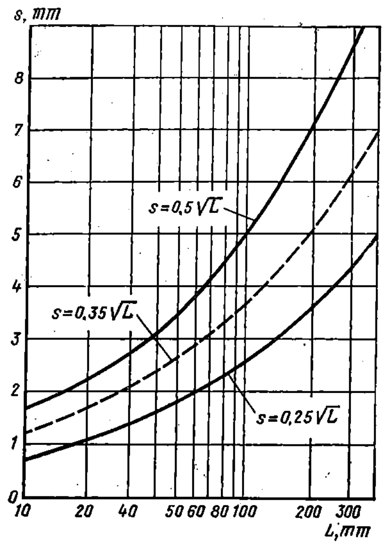


Fig. 442. Determining the mean thickness of walls for complex-shape moulded plastic parts

by ribs and walls should be smoothly filleted, and rib apexes should be rounded.

Dimensional relationships recommended for ribs are given in Fig. 443.

Inflection points on ribs and their junctions with walls should be filleted (Fig. 444).

As a general rule, internal ribs in parts should be preferred because the forming surfaces in the mould are relatively easy to machine on its convex part by disc-type form milling cutters. External ribs require machining in the concave part of the mould, which is very difficult and sometimes impossible. In some instances, the forming surfaces for external ribs can be made only with hand tools. The exception is forming surfaces for circular external ribs, which can be turned on a lathe.

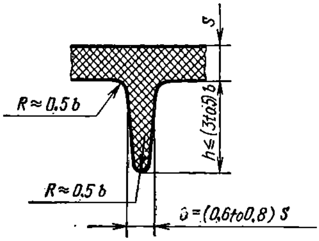


Fig. 443. Determining dimensions of ribs

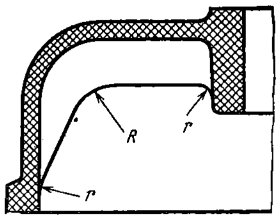


Fig. 444. Inflected rib

Straight ribs (in a plan view) should be preferred. Curved ribs require the forming surfaces in the mould that can be machined only by complex copying-milling methods.

When arranging ribs, it must be taken into account that, as a result of the part hardening and cooling, the ribs shrink and tend to distort the part removed from the mould by drawing together its walls. For this reason, ribbing should be avoided in parts where dimensional accuracy is vital, e.g. in gear wheels. Here, shrinkage of the ribs may cause waviness of the rim. If ribbing is necessary, small-height ribs of variable section (Fig. 445) having greater compliance are recommended for use.

Fillets. External and internal corners of parts should be rounded (Fig. 446). The radius of external corners should correspond to the diameter of end milling cutters used to machine the mould cavity, the minimum radius being $R = 2$-3 mm. The deeper the mould cavity, the greater should be the fillet radius; otherwise, the end milling cutter will not be adequately rigid, and hence the milling will be performed at low speeds and feeds.

To reduce the number of the tool types and sizes required, a single radius for all mould corners is recommended, or at least the number of different radii should be reduced to a minimum.

Mould cavities with sharp internal corners of the types shown in Fig. 447, *I-III* can be obtained only by broaching or slotting (for which a space for chips should be provided) or else by electrical discharge machining.

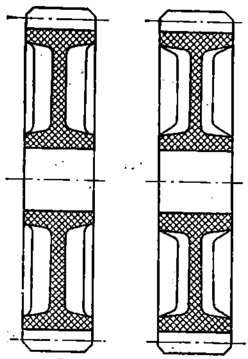


Fig. 445. Incorrect (*I*) and correct (*II*) design of ribbed parts

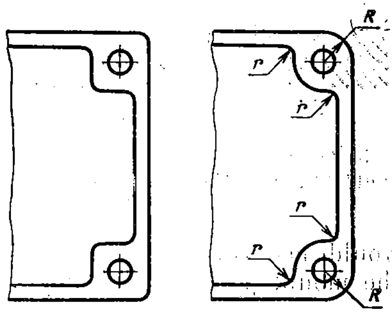


Fig. 446. Incorrect (*I*) and correct (*II*) moulding configurations

The internal corners of moulded parts are formed by the external corners of the mould convex parts, and therefore can be rounded with small radii (see Fig. 446) or even left sharp, since machining of the convex forming surfaces presents no difficulties. However,

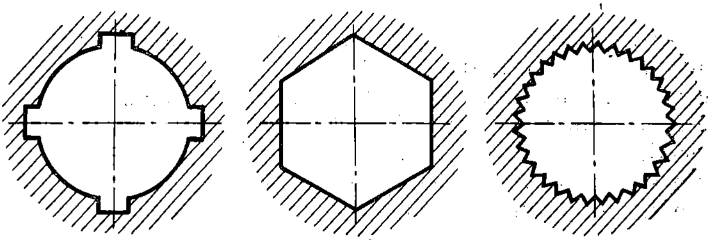


Fig. 447. Examples of difficult-to-make mould cavities

even here the maximum functionally permissible radii are recommended in order to ensure uniform filling of the mould and to reduce stress concentrations in the part due to shrinkage and working loads. The corner radius should be no less than 0.5 mm.

The junctions formed by the vertical and horizontal surfaces of the part should also be filleted. Here, the fillet radius determines the radius of the tip of a ball-nosed end milling cutter to be used for machining the bottom and side walls of the mould cavity.

The minimum radius for small parts is $R = 0.5$ mm; for medium-size and large parts, radii $R = 1$-3 mm are recommended. Greater bottom radii require end milling cutters of greater diameter, which will also result in greater radii of fillets between vertical walls.

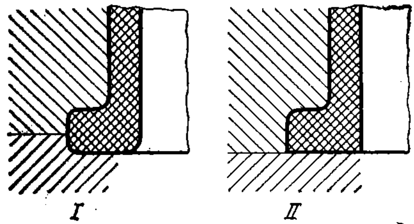


Fig. 448. Incorrect (*I*) and correct (*II*) shape of part surfaces formed at mould parting face

There are exceptions to the rule that external corners should be rounded. For instance, the corner formed by a vertical surface with a horizontal surface which lies in one plane with the mould parting face should not be rounded (Fig. 448). Otherwise, chamfering or rounding would necessitate the lowering of the horizontal surface below the mould parting face by the extent of the chamf r or the rounding radius. Rounding is also not used where its absence is specified by design requirements (Fig. 449).

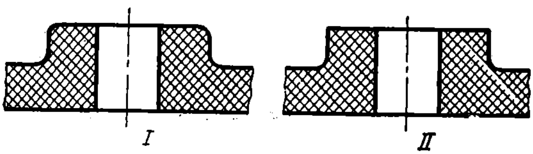


Fig. 449. Incorrect (*I*) and correct (*II*) shape of mating surface for subsequent assembly

Inscriptions and Signs. Inscriptions, signs, symbols, etc. made on outer surfaces of the moulded part should be embossed (Fig. 450). The forming surfaces for raised characters are relatively easy to machine in the mould, whereas machining embossed forming surfaces for sunk characters to be obtained on the product is very difficult indeed.

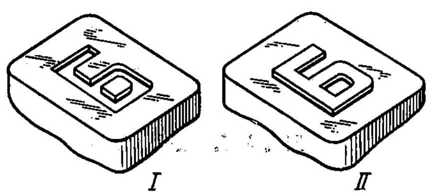


Fig. 450. Incorrect (*I*) and correct (*II*) lettering on plastic parts

Letters, etc., should be arranged on product surfaces which are parallel to the parting face of the mould. Raised characters on the lateral surfaces hinder the removal of the part from the mould.

Knurling. Hand-operated controls, threaded plugs and caps, etc. are usually provided with knurls for better grip by fingers. Examples of knurled parts are given in Fig. 451, *I-III*.

Knurls of different shape and size are found in practice. In order to simplify the manufacture of moulds for knurled parts, definite design rules should be followed. Knurls should be straight and parallel to the direction in which the part is ejected from the mould

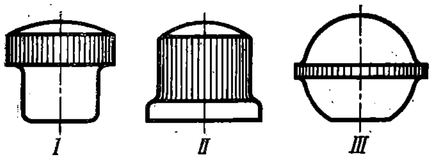


Fig. 451. Examples of knurled plastics parts

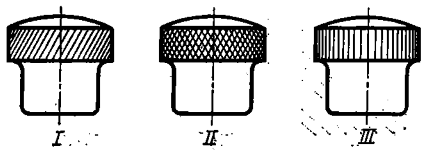


Fig. 452. Incorrect (*I*, *II*) and correct (*III*) knurling

(Fig. 452, *III*). Diagonal and diamond type knurls (Fig. 452, *I* and *II*) cannot be used as they are hard to machine on the mould surfaces and hinder the removal of the part from the mould.

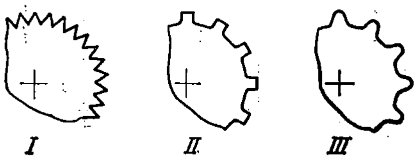


Fig. 453. Incorrect (*I*, *II*) and correct (*III*) knurl profiles

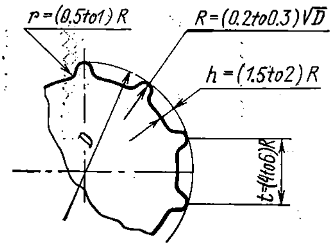


Fig. 454. Determining dimensions of knurl profile

The cross-sectional profile of knurls should be rounded at the apex (Fig. 453, *III*) to simplify machining of the forming surfaces in the mould.

The pitch of knurls should be as large as possible for convenient handling of the moulded part. The recommended shape and dimensions of knurls are given in Fig. 454.

6.3.2. Inserts

Plastic parts are often provided with metal bushings, threaded inserts, rods, pins, etc. The inserts placed in the mould are firmly embedded in the plastic material as this is moulded and solidified.

Inserts should be used only where indispensable, because they complicate the mould and lower production rates.

A number of requirements must be met when introducing inserts in plastic parts. These elements should be fixed within the mould

against both axial and transverse displacements. Rod and pin type inserts are located in special holes of the mould. Short rods are fixed axially by being brought to bear against the bottoms of their holes (Fig. 455, *I*). Long rods extending beyond the mould are shouldered against the counterbores of their holes (Fig. 455, *II* and *III*).

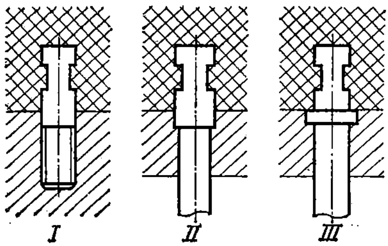


Fig. 455. Locating inserts in moulds

Bushing-type inserts are fixed on pins entering their holes. Tapped inserts are centred by pins on the thread minor diameter. Bushings extending out of the moulded part can be fixed in suitable holes of the mould.

Some difficulties may be experienced if inserts have to be fixed in the upper half of the mould, wherefrom they can slip under gravity.

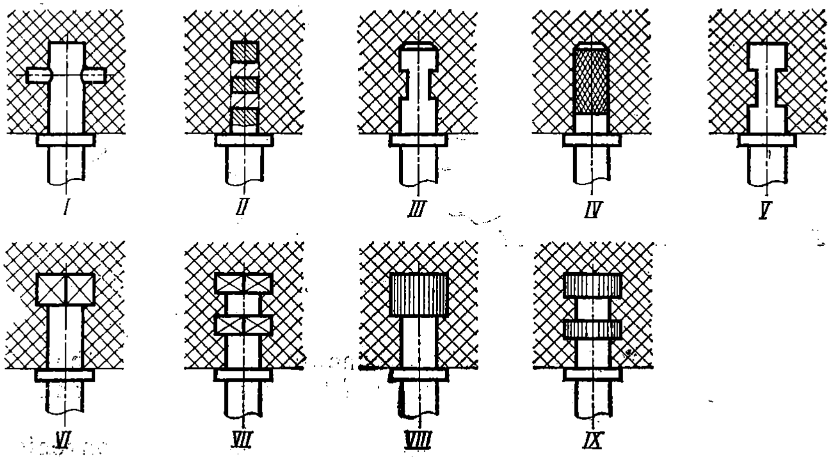


Fig. 456. Anchoring inserts in plastics parts

To prevent this, spring-actuated locking devices are resorted to. The arrangement of inserts in the upper half of the mould is not recommended.

These difficulties are largely obviated in the moulds with a vertical parting face. But here measures must be taken to prevent the inserts from casually shifting off their locating elements.

As the mould is opened, the inserts must freely leave their locating holes or other features. The inserts axes should be perpendicular to the mould parting face. If they are parallel or inclined, the mould

construction becomes very complicated, requiring retractable or removable locking elements.

With rod inserts, free insertion of the rod into the locating hole and free removal therefrom during ejection of the moulding must be provided for. The free end of the rod must be smaller in diameter than the locating hole. The use of bent rods is impermissible. If necessary, rods should be bent after moulding.

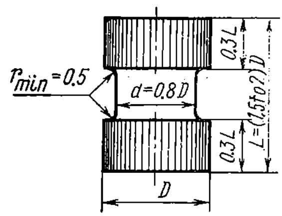


Fig. 457. Determining dimensions of knurled collars for insert anchoring

Very long rod inserts are undesirable. Such rods are better to assemble to plastic mouldings, e.g. by screwing them into a threaded insert in the moulded part.

Inserts should be securely anchored in mouldings against axial and angular displacements. Various rod-type inserts are shown in Fig. 456, *I-IX*. The most commonly used inserts are those with knurled collars separated by a circular neck (Fig. 456, *IX*) Dimensional relationships for such inserts are shown in Fig. 457. A simpler form

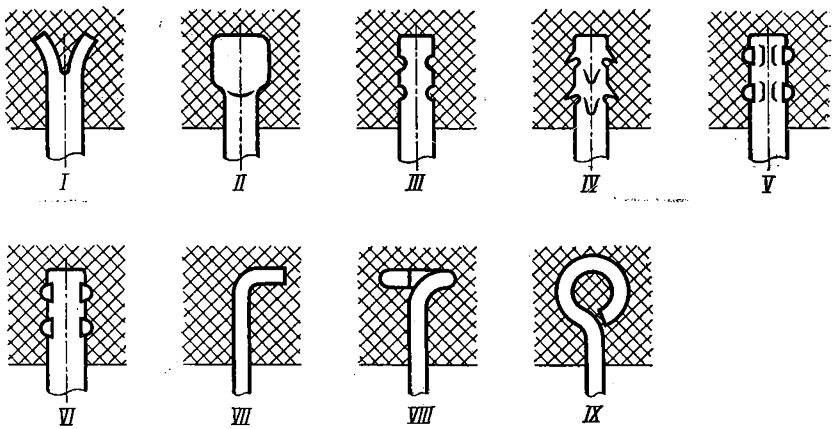


Fig. 458. Rod inserts embedded in plastics parts

of insert, a rod with two flats milled thereon for good anchorage (Fig. 456, *V*), is also often used.

Shown in Fig. 458, *I-IX* are inserts with moulded-in ends preformed by plastic deformation methods. Such inserts are simpler and quicker to produce than the inserts of Fig. 456, which require machining operations. The latter methods, however, can only be applied to rods of plastic metals.

Of all the inserts shown in Fig. 458, the most expedient is that where wings are pressed on the end (Fig. 458, *VI*) using a simple and quick die forging operation.

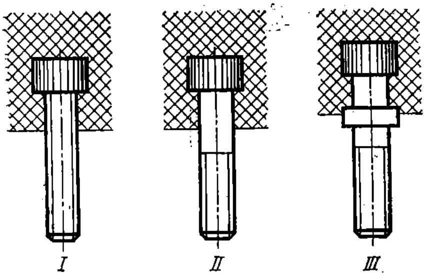


Fig. 459. Incorrect (*I*) and correct (*II*, *III*) setting of threaded rod inserts

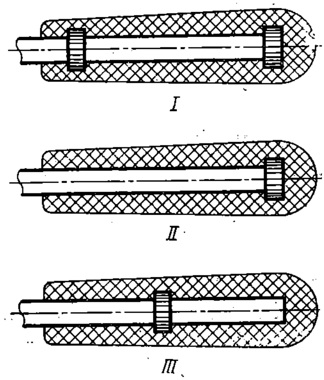


Fig. 460. Incorrect (*I*) and correct (*II*, *III*) methods of anchoring long rod inserts

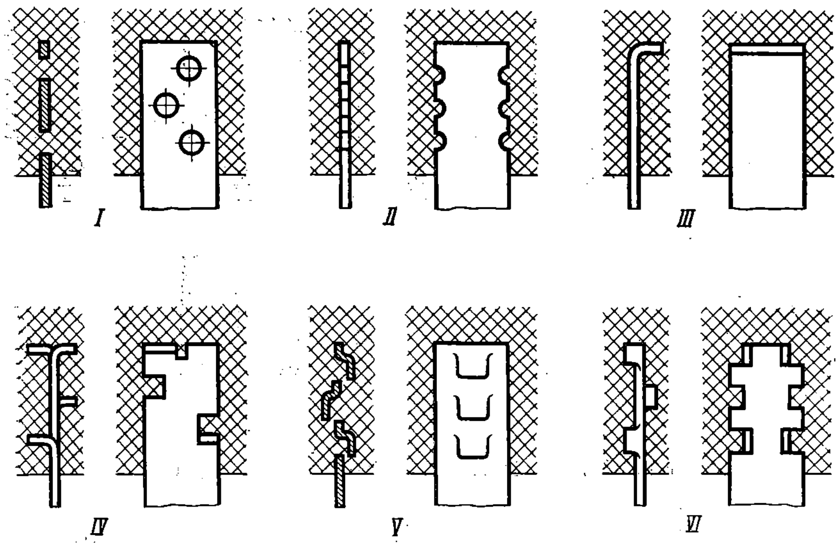


Fig. 461. Sheet metal inserts embedded in plastics parts

When using threaded rod inserts, it is advisable to avoid the embedding of the threads in the moulded part (Fig. 459, *I*). Where it comes out of the part, a threaded rod should be plain (Fig. 459, *II*), or, better still, have a collar (Fig. 459, *III*).

The anchoring elements of an insert should not be spaced far apart, as i Fig. 460, *I*, where a long plastic handle is shown moulded around

Fig. 462. Sheet metal inserts set parallel to moulding face

a metal rod. The shrinkage of the plastic material while it is hardening develops stresses in the region between the collars of the insert, which may cause cracking. For this reason, a single collar is preferred

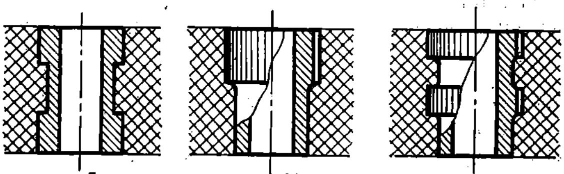


Fig. 463. Bushing-type inserts in moulded plastic parts

(Fig. 460, *II* and *III*), which allows the material to shrink along the insert's plain portions

Fig. 461, *I-VI* shows sheet metal inserts moulded in plastic parts.

Sheet metal inserts disposed parallel to the part face are shown in Fig. 462, *I-IV*. Bushing-type inserts are most commonly fixed by the

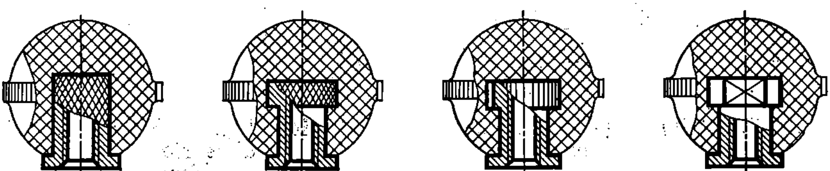


Fig. 464. Anchoring tapped inserted in moulded ball grips

methods illustrated in Fig. 463, *I-III*. Tapped inserts moulded in ball grips are shown in Fig. 464, *I-IV*.

Where bushing-type inserts are to be moulded in blind holes, the insert hole itself should be blind (Figs. 465, *II*, 466, *II*). Otherwise (Figs. 465, *I*, 466, *I*), the plastic material will flow into the hole in the process of moulding.

The setting of tapped inserts flush with the part face (Fig. 467, *I*) should be avoided, since the plastic material may flow during mould-

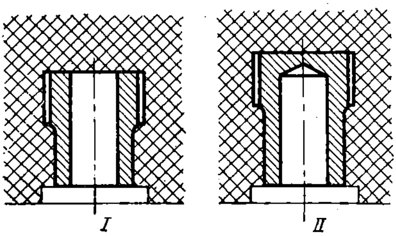


Fig. 465. Inserts for blind holes
I—incorrect; *II*—correct

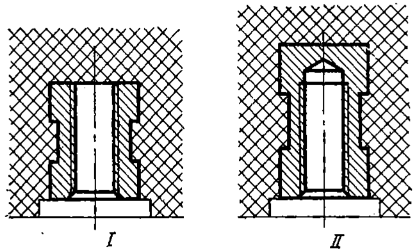


Fig. 466. Inserts for blind tapped holes
I—incorrect; *II*—correct

ing under the face of the part and form a film, which subsequently has to be removed.

Better arrangements are either to sink the insert by 2 to 3 mm into the part (Fig. 467, *II*) or to make it extend by the same amount

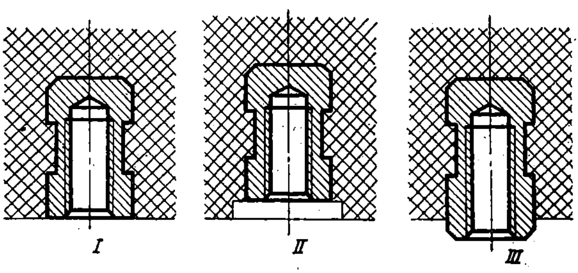


Fig. 467. Setting tapped inserts in moulded plastic parts
I—improper; *II*—proper

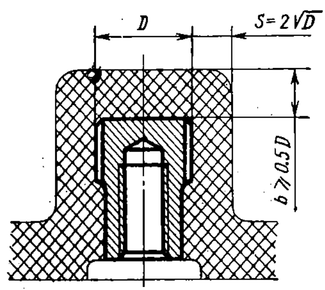


Fig. 468. Determining dimensions of boss in moulding to accommodate insert

(Fig. 467, *III*). The formation of the plastic film is prevented by the cylindrical collar on a pilot pin in the first instance, and on the projecting portion of the insert in the second.

The walls of a boss with a metal insert shoud be thick enough to exclude its cracking from shrinkage. The wall thickness can be found from an empirical formula: $s = 2\sqrt{D}$, where D is the insert diameter in mm (see Fig. 468).

6.3.3. General Design Rules

Design of plastic parts should take account of not only the processing factors but also specific mechanical properties of plastics, such as low hardness, stiffness, and strength, and, with thermosets, increased brittleness.

Low stiffness of plastics should be compensated for by ribbing (Fig. 469), by introducing flanges and beads (Fig. 470, *I*, *II*) and

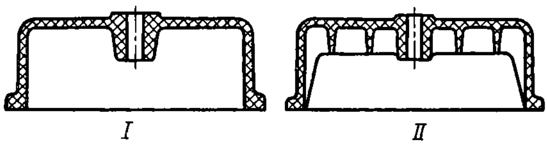


Fig. 469. Incorrect (*I*) and correct (*II*) design of moulding

by providing a part with a stiff ribbed vault (Fig. 471). The ribbed vaults of parts subjected to heating during service make for reduced stresses caused by thermal distortions, which may be significant owing to the great linear expansion coefficients of plastics.

Care should be taken to avoid loading plastic parts with bending forces; they should be loaded with more favourable compression

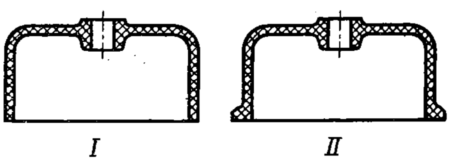


Fig. 470. Strengthening moulded part with a bead

forces (Fig. 472, *I* and *II*). Bending forces due to tightening of fasteners are impermissible (Figs. 473 and 474). To prevent damage to bearing surfaces on plastic parts, large-diameter washers should be placed under the heads of fasteners (Fig. 475, *II*), or, better, inserts should be introduced into the clearance holes (Fig. 475, *III*).

Fastening plastics parts with slotted countersunk head screws (Fig. 476, *I*) is undesirable as these screws, when tightened, give rise to tensile breaking stresses in the hole walls. In addition, the part surface around the countersink can be easily damaged by a screwdriver.

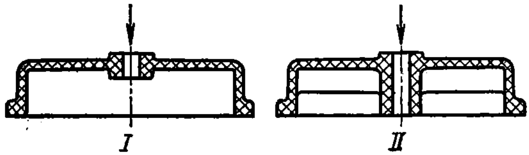


Fig. 471. Incorrect (*I*) and correct (*II*) design of moulding

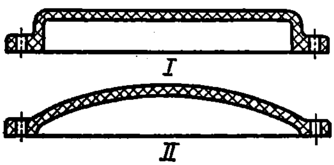


Fig. 472. Moulding shape modified for compressive loading

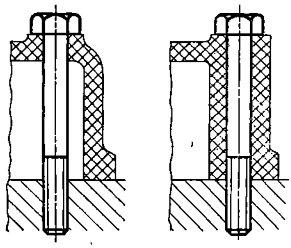


Fig. 473. Eliminating bending loads caused by screw tightening

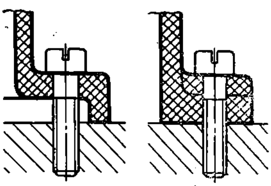


Fig. 474. Flange shape modified to eliminate bending due to screw tightening

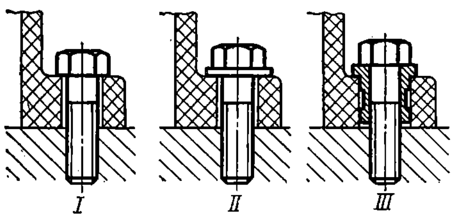


Fig. 475. Preventing bearing surfaces of moulded parts from crushing by screw heads

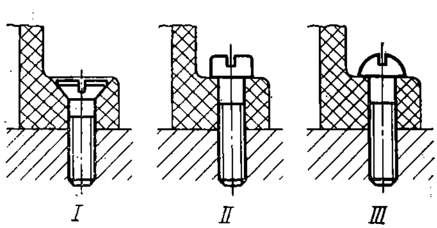


Fig. 476. Improper (*I*) and proper (*II*, *III*) types of screws for fastening moulded parts

6.4. Plastic Gear Wheels

Gear wheels made of plastics (to mesh with metallic pinions) are used in medium-loaded transmissions (e.g. auxiliary drives, instruments, etc.). Transmissions with plastic gears run smoothly and quietly and (when rationally designed) have long service life. Plastic gears can operate with little oiling, and at low speeds and loads, without any oiling at all.

Low strength and hardness of plastics disqualify them for highly loaded gear transmissions.

Approximate calculations of plastic gears can be based upon the condition that a load for 1 cm of tooth width should not exceed 20

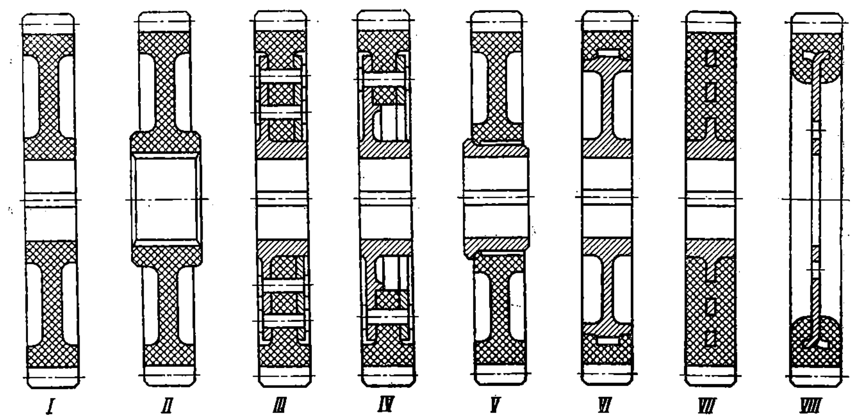


Fig. 477. Plastic gear wheels

to 30 kgf/cm for reinforced plastics (cloth and wood laminates) and 15 to 20 kgf/cm for plastics without fillers. (By comparison, the specific load for steel gears with hardened or nitrided teeth can be as high as 200 kgf/cm, and in some instances, 500 kgf/cm.) Low heat conduction of plastics, which impedes the removal of heat from the surfaces of friction, must be taken into account as well as their low strength.

Consequently, the use of plastic gears in high-power transmissions is possible only at low rotational speeds in low-speed transmissions using large-diameter gears with increased face width.

Gear wheels are made mainly of cloth laminates, wood laminates, capron, nylon, and polyformaldehydes. Gears of cloth and wood laminates are obtained by machining, those of capron, nylon, and polyformaldehydes are injection moulded.

Gears of cloth and wood laminates provide the necessary strength and long life if the layers of cloth (or wood) in the material are disposed at right angles to the teeth flanks.

Fig. 477 shows plastic gears in various forms. Gear wheels are mounted on shafts with the use of a key for torque transmission (Fig. 477, *I*) only where loads are insignificant, because otherwise the key seat may get crushed. Greater loads require wheel hubs of increased length and diameter and splines (Fig. 477, *II*) rather then keys.

A stronger and more reliable support is provided by flanged steel hubs (Fig. 477, *III* and *IV*), with plastic gear reams fastened to the flange by rivets or screws. Metal plate discs must be placed under the rivet heads (or under the screw heads and nuts).

Moulded gears are used with steel hub inserts (Fig. 477, *V*). High strength and reliability are provided by the design shown in Fig. 477, *VI*, where the gear ream is pressed on a large-diameter disc hub with a fluted periphery.

Fig. 477, *VII* shows a gear ream moulded around a disc-type perforated hub. Gear wheels for low loads (Fig. 477, *VIII*) are obtained by moulding the gear rim over a disc which is then fastened to the shaft with bolts. The disc is anchored in the rim by various methods, e.g. by claws bent on the disc.

Typical Assemblies

7.1. Fixing Pivot-Type Components

The fixing of pivots, axles, pins, rods, bars and similar cylindrical parts is a task very frequently encountered in machine design.

Depending on its function, a part of this type must be fixed against either axial or angular displacement or both.

The most commonly used methods for fixing are considered below. The fixed part is assumed to be mounted in two supports.

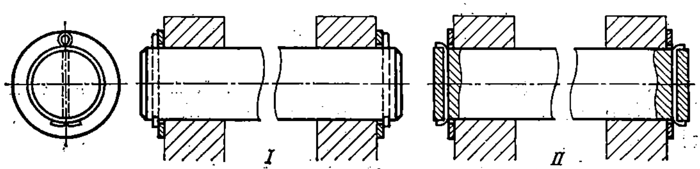


Fig. 478. Pivot pin fixed against axial displacement by washers and cotter pins

The simplest methods of axial locking are effected by means of washers with cotter pins placed at the pivot ends (Fig. 478, *I*, *II*). Such methods are unreliable since the cotter pins may break down under axial loads. In this arrangement, clearances should be provided between the washers and the supports' outward end faces to compensate for possible manufacturing inaccuracies and temperature distortions.

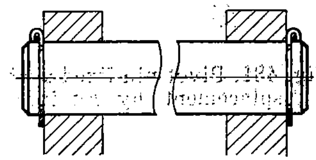


Fig. 479. Fixing against axial displacement by ШЕЗ washers

A more reliable fixing method by means of ШЕЗ washers (Egorov's lock washers) is illustrated in Fig. 479. Such a washer is a soft-steel form stamping *a* (Fig. 480) inserted into a groove on the end of a joint pin. The overhanging part of the washer is then bent back to bear against the pin end, whereby the washer is fixed on the pin.

Locking against axial displacement can be effected by retaining rings (Figs. 481-484). These can be placed at the ends of the pin, as shown in Fig. 481, sometimes with washers placed in counterbores on the supports faces (Fig. 482). An assembly where the pin has a

head at one end and carries a retaining ring on the other is shown in Fig. 483. The assembly of Fig. 484 makes use of retaining rings inserted in the bores of the supports. In yet another variation (Fig. 485),

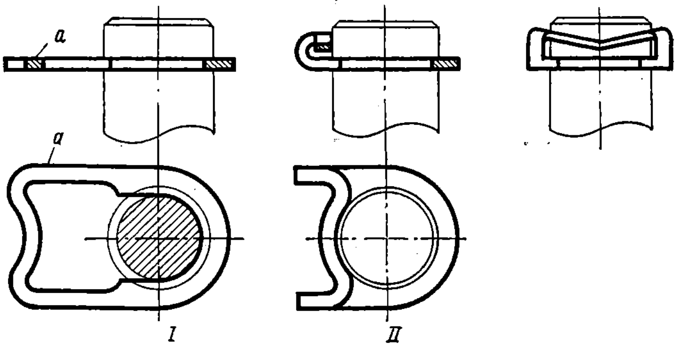


Fig. 480. Setting a ШЕЗ washer
I—first stage; *II*—second stage

the pin has a deep groove for a retaining ring. During assembly, the retaining ring snaps into a groove in the bore and locks the pin.

In the assemblies shown in Figs. 486 and 487, the pin is fixed against axial displacement by radially inserted retaining rings placed on the inner side of the supports.

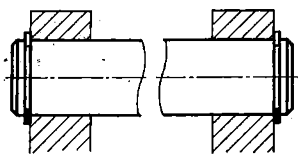


Fig. 481. Pivot pin fixed against axial displacement by retaining rings

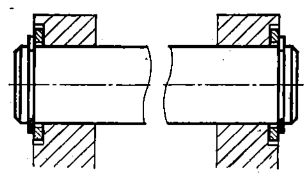


Fig. 482. Pivot fixed against axial displacement by washers and retaining rings

Fig. 488 shows a pin fixed against axial displacement by a retaining ring and a lid bolted to one of the supports. In a variation of the latter example (Fig. 489) the pin is also locked against rotation, for which purpose it is provided with flats that engage a correspondingly shaped hole in the lid.

A method for securing a pin with a wire placed in grooves of half-round section on the pin and in the hole is shown in Fig. 490.

The pin in Fig. 491, *I* is locked by plastically deformable thin spherical discs; these are fixed in the counterbores of the supports by flattening their spheres. A variation of this locking is shown in

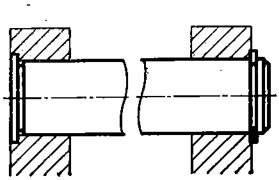


Fig. 483. Pivot pin fixed against axial displacement by its shoulder and retaining ring

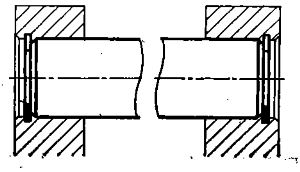


Fig. 484. Pivot pin fixed against axial displacement by retaining rings placed in its bores

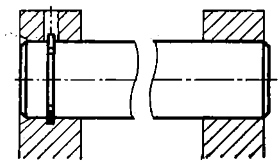


Fig. 485. Pivot pin fixed against axial displacement by snap ring

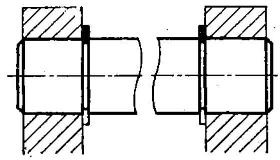


Fig. 486. Pivot pin fixed against axial displacement by radially set retaining rings

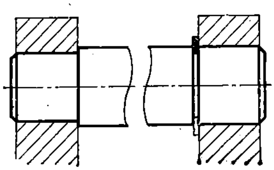


Fig. 487. Stepped pin fixed against axial displacement by radially set retaining ring

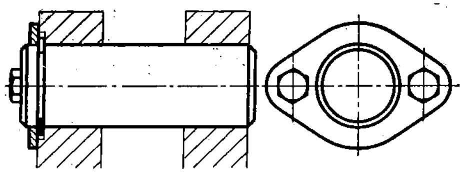


Fig. 488. Pivot pin fixed against axial displacement by bolted end plate and retaining ring

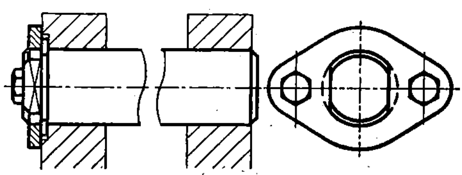


Fig. 489. Pivot pin fixed against axial and angular displacements by end plate with contoured hole and retaining ring

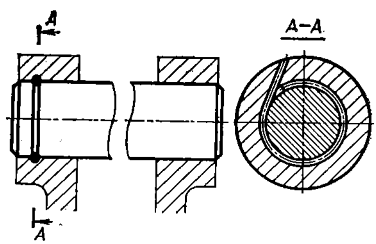


Fig. 490. Pivot pin fixed against axial displacement by a length of wire passed along half-round grooves on the pin and in its bore

Fig. 491, *II*. Here, a spherical washer is flattened by tightening the screw; as this takes place the washer is forced into a groove provided in one of the supports.

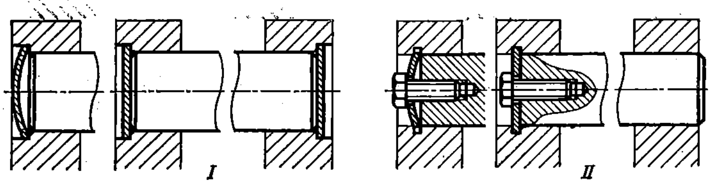


Fig. 491. Pivot pin fixed against axial displacement by flattening deformable spherical washers

A pin can be pressed by its knurled end into the hole in one of the supports (Fig. 492). This method is used where the supports are made of a plastic material.

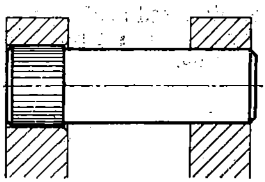


Fig. 492. Pivot pin press-fitted by its knurled end

Fig. 493, *I*, *II* illustrates the locking of a joint pin by end washers held by screw fasteners.

The pin shown in Fig. 494 is fixed against axial displacement by two flanges bolted to the supports. One of the flanges also locks the pin against rotation by its tenon which fits into a slot on the pin end face. Fig. 495 illustrates a method whereby a pin is fixed in position with the aid of several balls placed in holes made in the pin along its periphery. As the screw is tightened, the conical insert pushes the balls radially into a circular groove in the bore. Locking against rotation is effected by friction.

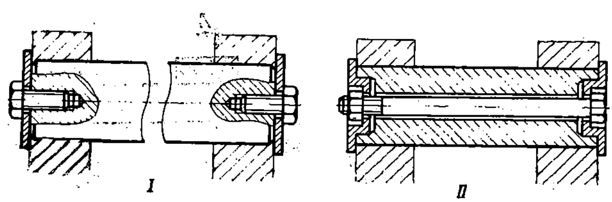


Fig. 493. Pivot pin fixed against axial displacement by end washers

A permanent assembly is shown in Fig. 496. Here, the pin is fixed against angular and axial displacements by a set screw placed at the junction of the pin and bore mating surfaces.

Locking with cylindrical and taper fastening pins (Figs. 497 and 498, respectively) is inconvenient for manufacturing reasons, since

it requires drilling and reaming with the joint pin in position. Somewhat better in this respect is the assembly shown in Fig. 499, where a cylindrical pin is placed into the slot on the joint pin end face.

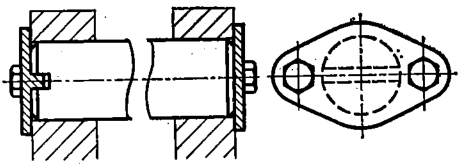


Fig. 494. Pivot pin fixed against axial and angular displacements by end plates, one of these having tenon to prevent pin rotation

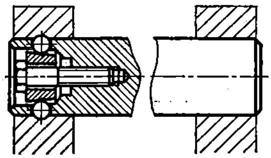


Fig. 495. Pivot pin fixed against axial displacement by balls

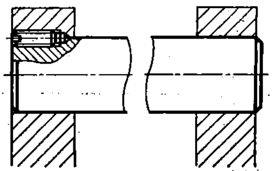


Fig. 496. Pivot pin fixed by set screw

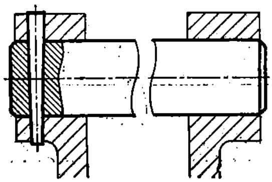


Fig. 497. Pivot pin fixed by cylindrical pin

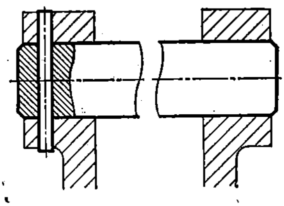


Fig. 498. Pivot pin fixed by taper pin

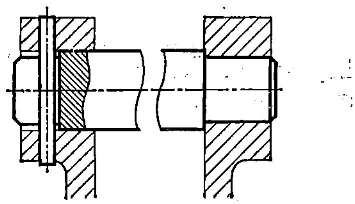


Fig. 499. Pivot pin fixed by cylindrical pin engaging its end slot

Fixing with set screws is illustrated in Figs. 500-504. Set screws with cone points (Fig. 502) provide the most reliable locking against axial and angular displacements without any clearances and play. Set screws can be installed to engage circular grooves on the pins (Figs. 503 and 504) when the latter require angular adjustment. The pin is fixed in position by friction; set screws with cone points (Fig. 504) provide a fairly reliable locking.

A pin is very convenient to fix against axial displacement where it rotates in supports and carries a hub rigidly secured thereon so

that the hub is located between these supports (Fig. 505, *I*, *II*). Here, the pin is locked through the hub.

Pivot pins can be fixed at one end against both axial and angular displacements (Figs. 506-510). The fixing is effected by a circular nut

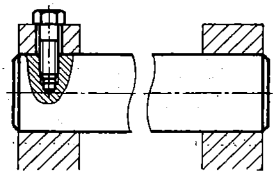


Fig. 500. Pivot pin fastened by screw

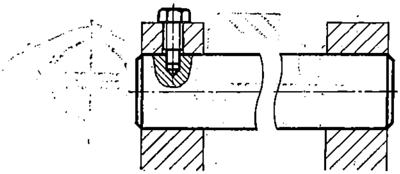


Fig. 501. Pivot pin fixed against axial and angular displacements by set screw with dog point

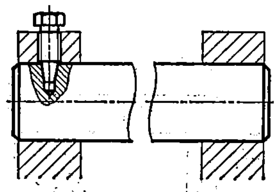


Fig. 502. Pivot pin fixed against axial and angular displacements by set screw with cone point

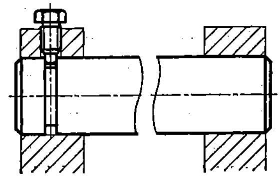


Fig. 503. Pivot pin fixed against axial displacement by set screw with dog point

and key (Fig. 506); by a screw and a pin (Fig. 507); by a screw and a key which engages a slot on the pivot-pin end face, the key being made integral with the flange bolted to the housing (Fig. 508); by a circular nut tightened on the tapered end of the pivot pin, without special means for locking against rotation, which are unnecessary in most of such cases (Fig. 509); and by a spring clamp (Fig. 510).

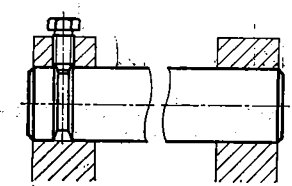


Fig. 504. Pivot pin fixed against axial displacement by set screw with cone point

Locking by a bolt fitting into a circular groove on the pin is shown in Fig. 511, and by a bolt with an arcuous recess for contact with the pin, in Fig. 512. Reliable locking is provided by tightening a bolt with a wedge in the form of a flat on its shank as shown in Fig. 513; here the pin is fixed against angular and axial displacements.

A peculiar method is shown in Fig. 514, *I*, *II*. The pins are mounted on top of the supports and fastened there. The method is applied when

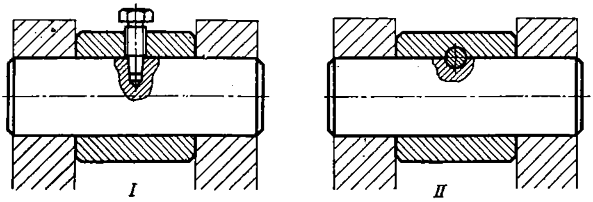


Fig. 505. Rotatable pin fixed against axial displacement by hub secured thereon

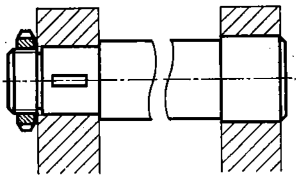


Fig. 506. Pivot pin fixed against axial and angular displacements by circular nut

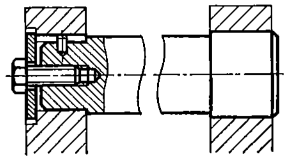


Fig. 507. Pivot pin fixed agains axial and angular displacements by cap screw and washer

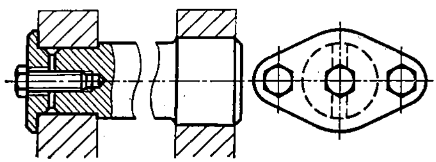


Fig. 508. Pivot pin fixed agains axial and angular displacements by flange with tenon

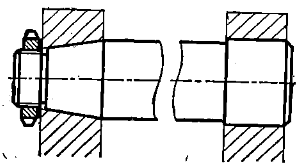


Fig. 509. Pivot pin fixed against axial and angular displacements by tightening its tapered end with circular nut

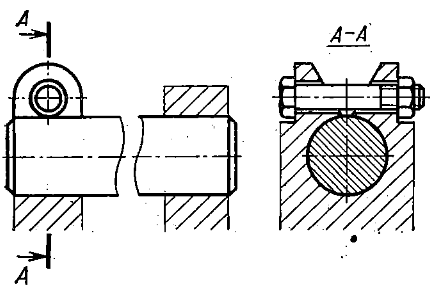


Fig. 510. Fastening by spring clamp

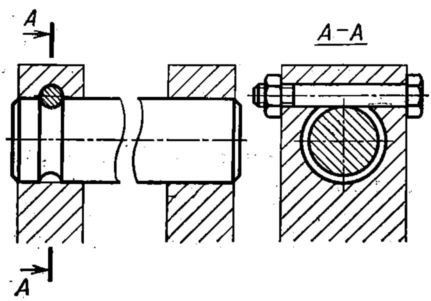


Fig. 511. Pivot pin fixed against axial displacement by bolt engaging a groove of semi-circular profile on the pin

accurate location of the pins relative to the supports is not required and the distance between the supports is small (with greater distances, distortions due to temperature changes are possible).

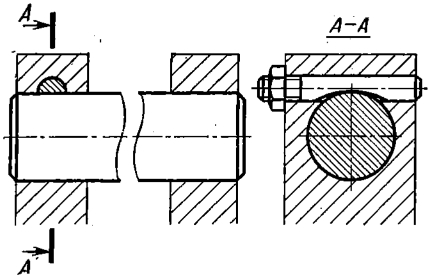


Fig. 512. Pivot pin clamped by bolt with cylindrical recess

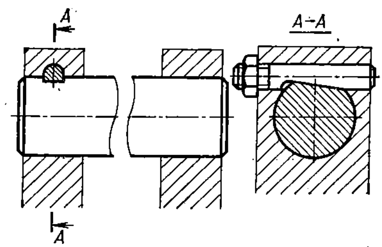


Fig. 513. Pivot pin fixed against axial and angular displacement by wedge bolt

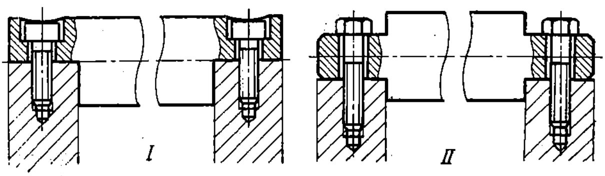


Fig. 514. Pivot pins fastened with cap screws

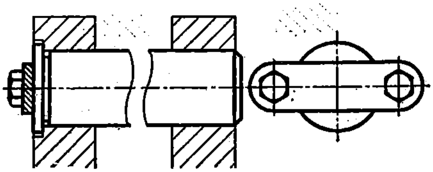


Fig. 515. Pivot pin fixed against axial displacement by its shoulder and end plate

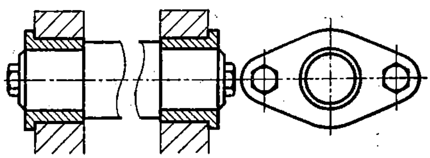


Fig. 516. Pivot pin fixed against axial displacement by bushings secured in supports by screws

In the assembly of Fig. 515, the pin is fixed axially by its head and a bolted end plate.

Assemblies with bolted bushings are shown in Figs. 516-517. In Fig. 518 is shown a pin with a flange-type head which is fastened to the housing. This arrangement provides reliable fixing against axial and angular displacements; the drawback is a more complex shape of the pin.

The intent to have a straight pin has resulted in various designs with tongue-type locking elements. These are exemplified by a washer fastened to one of the supports (Figs. 519 and 520). Figs. 521,

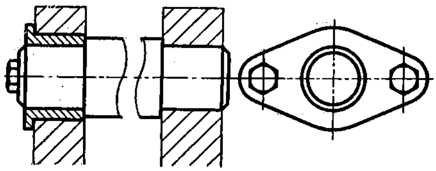


Fig. 517. Stepped pivot pin fixed against axial displacement by a single bushing

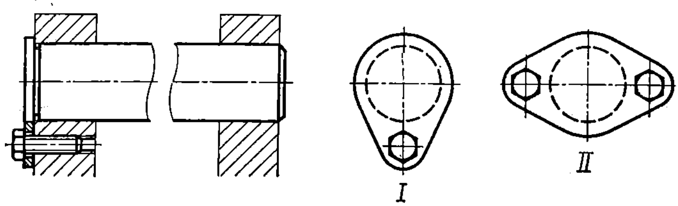


Fig. 518. Pivot pin fixed against axial and angular displacement by fastening its flange with screws (*I* and *II* — flange variants)

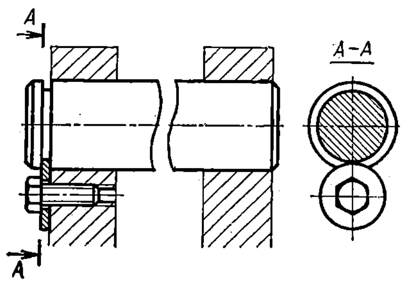


Fig. 519. Pivot pin fixed against axial and angular displacements by screw and washer

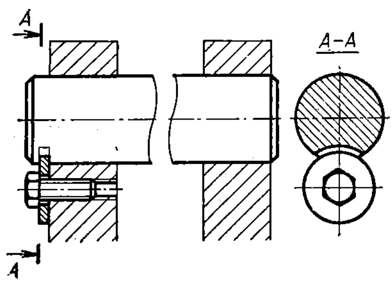


Fig. 520. Pivot pin fixed against axial and angular displacements by screw and washer

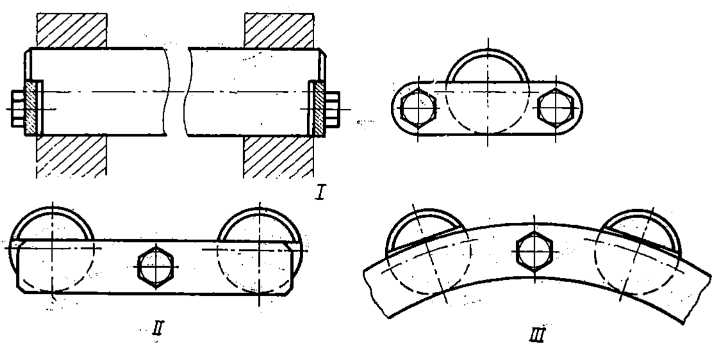


Fig. 521. Pivot pin fixed against axial and angular displacements by end plates

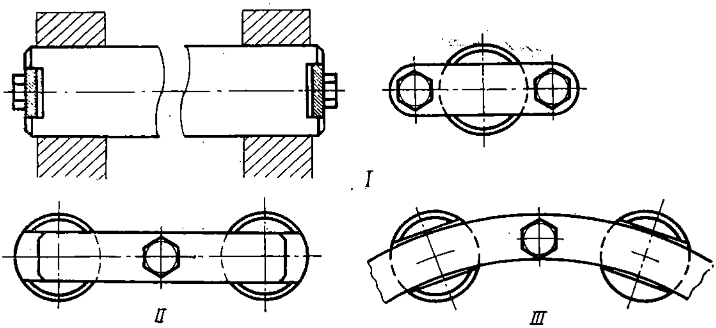


Fig. 522. Pivot pin fixed against axial and angular displacements by end plates

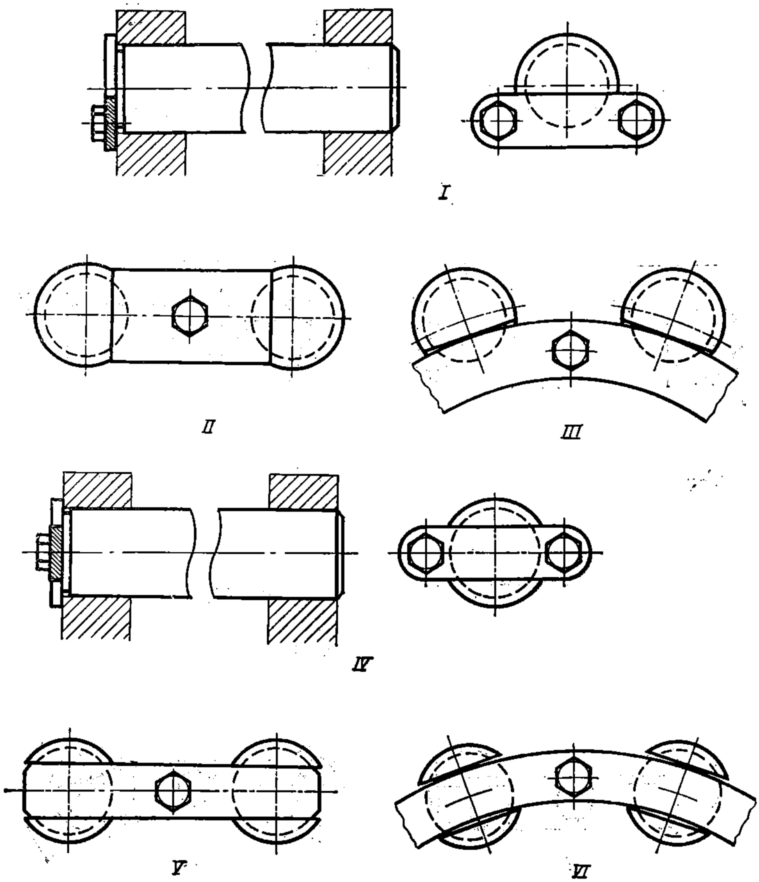


Fig. 523. Shouldered pivot pins fixed against axial and angular displacements by end plates

I-III and 522, *I-III* illustrate methods of axial and angular location by means of end plates bolted to the supports at both ends of the pin.

These methods are well adapted for multiple-pin locking. Figs. 521, *III* and 522, *III* give examples of such methods as applied

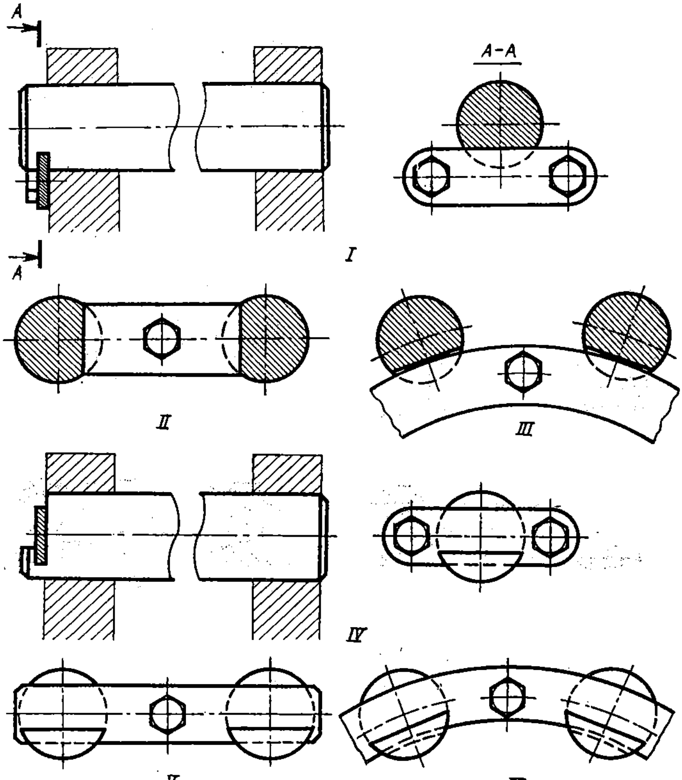


Fig. 524. Pivot pins fixed against axial and angular displacements by end plates

to the fixing of a number of pins disposed in a circle. Here, the end plates take the shape of rings.

Fixing by a single plate disposed at one end of the pin (Figs. 523-526) is a more advanced method. In the assemblies shown in Fig. 523, *I-VI*, the plates are sunk into the slots on the pin end faces. The pin is fixed against axial displacement as its head is captured between the end plate and the support face.

Figs. 524-528 illustrate methods for securing straight pins. In the assemblies shown in Fig. 524, *I-VI*, the end plates are inserted in transverse slots made on the pins. Fixing a number of pins located

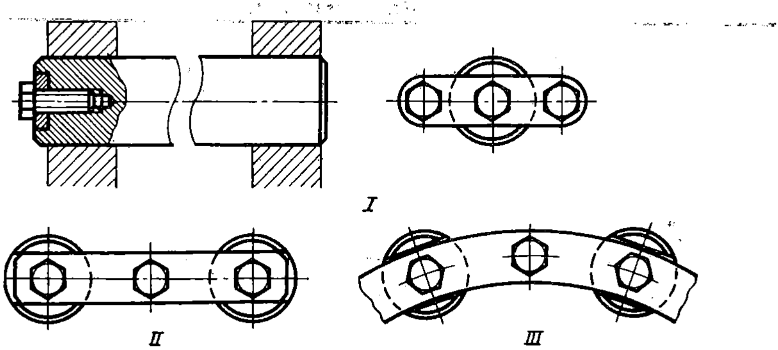


Fig. 525. Pivot pins fixed against axial and angular displacements by end plates

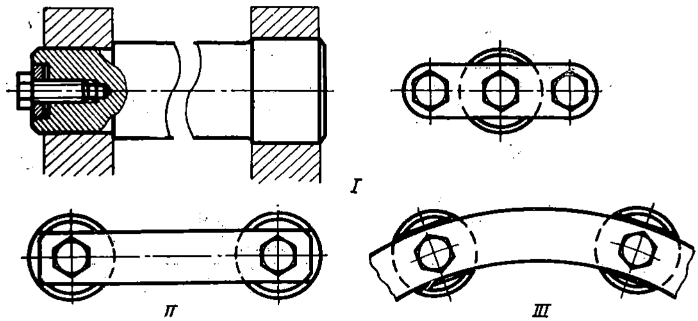


Fig. 526. Stepped pivot pins fixed against axial and angular displacements by end plates

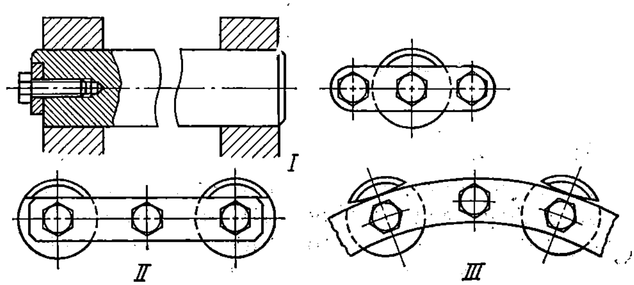


Fig. 527. Pivot pins fixed against axial and angular displacements by end plates

in a circle requires that the locking ring should consist of two half-rings.

In the assemblies shown in Figs. 525-528, the end plates are fitted into recesses made on the pin end faces. A stepped pin (Fig. 526, *I*,

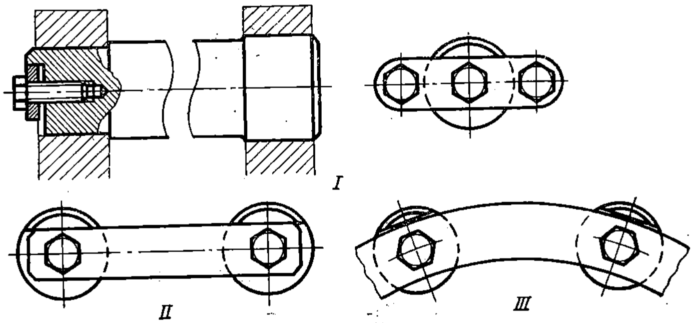


Fig. 528. Stepped pivot pins fixed against axial and angular displacements by end plates

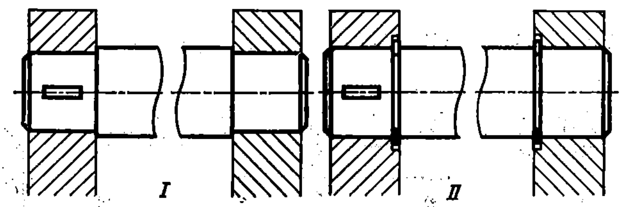


Fig. 529. Pivot pins fixed against axial and angular displacements between detachable supports

II, *III* and Fig. 528, *I*, *II*, *III*) can be secured by firm screw tightening. A number of circularly positioned pins of this type can be secured by a whole end ring.

The pins in Figs. 527, *I*, *II*, *III* and 528, *I*, *II*, *III* have recesses which are favourably simpler in shape than those of the pins in Figs. 525, *I*, *II*, *III* and 526, *I*, *II*, *III*.

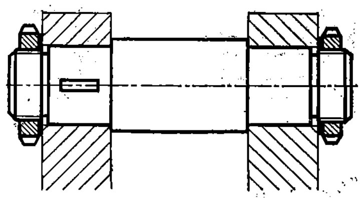


Fig. 530. Pivot pin fixed against axial and angular displacements between detachable supports and used to hold supports together

Some methods for securing pins in separable supports are shown in Figs. 529 and 530. Where the supports are held together by fasteners, the pin can be fixed by its shoulders as in Fig. 529, *I*, or by retaining rings as in Fig. 529, *II*.

Fig. 531, *I-XII* illustrates methods whereby pins are secured in small shackles, brackets, yokes, etc.

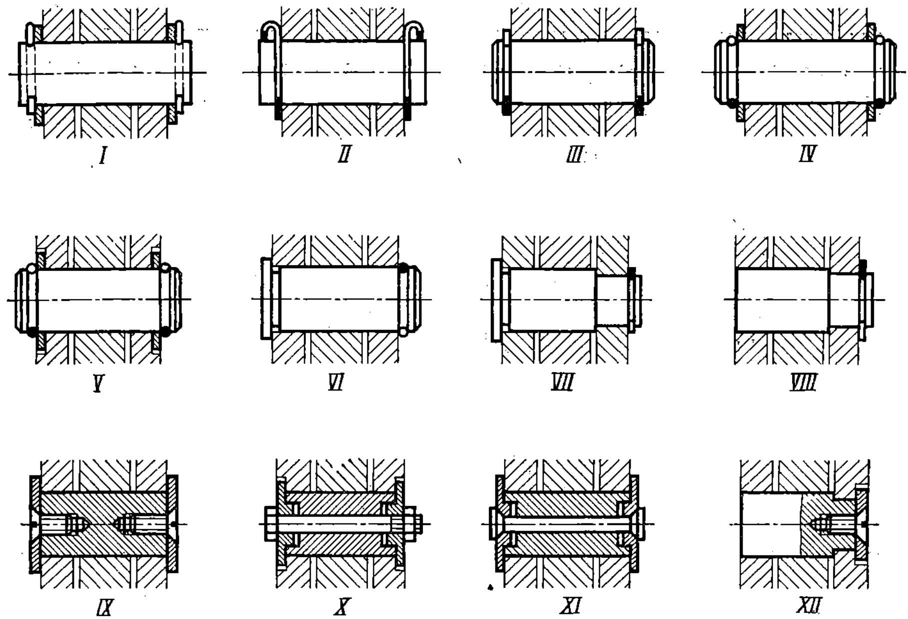


Fig. 531. Small pivot pins in separable pin joints

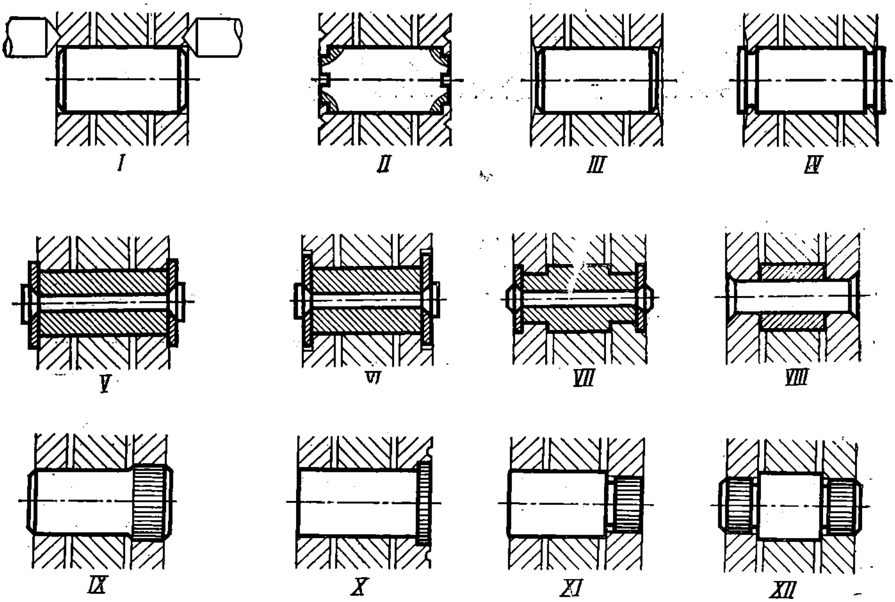


Fig. 532. Small pivot pins in permanent pin joints

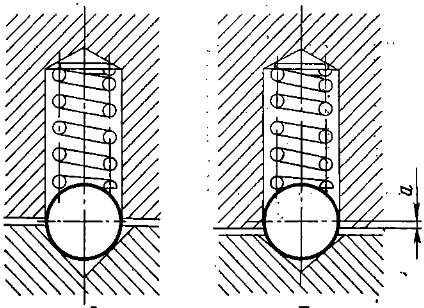
Methods for fixing hardened pins in supports of soft steel are presented in Fig. 532. The pins are fixed by staking (Fig. 532, *I*, *II*), by fluting (Fig. 532, *III*, *IV*), by riveting end washers (Fig. 532, *V-VII*), by using rivets (Fig. 532, *VIII*), and by pressing into the supports their knurled ends (Fig. 532, *IX-XII*).

7.2. Locking Devices

These are used to fix temporarily a part which rotates or moves in straight guide ways relative to another part.

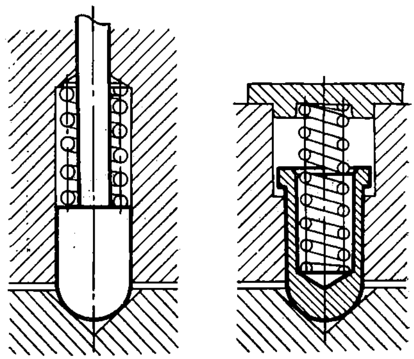
The locking may be stepless, i.e. the movable part can be locked in any position, or stepped, i.e. the part is locked in definite positions.

The locking may be elastic or positive. In the first case, the locking device retains the part with a definite (usually small) force. To move



I *II*

Fig. 533. Spring-loaded ball as a locking device
I—incorrect; *II*—correct



I *II*

Fig. 534. Cylindrical spring plungers with spherical points

the part into another position, this force must be overcome. In the second case, the locking device engages in associated seats on the stationary part and thereby rigidly fixes the movable part. To move the latter into another position, the locking device must be withdrawn from the seat.

A locking device in its simplest form is a spring-loaded ball placed in a cylindrical bore in one of the parts (Fig. 533); it is used to effect elastic locking. Under the pressure of the spring, the ball gets into a suitable seat provided in the other part. The force that keeps the ball in place is proportional to the spring's compression and the inclination of the seat walls. To move the part into another position, it is necessary to apply a force in the direction of movement which is sufficient to compress the spring and force the ball out of its seat.

The spring-loaded ball as a locking device suffers from certain disadvantages. To prevent its jamming, the ball should be sunk into the seat in such a way that its centre in the extreme lower position remains above the edges of the seat by amount a (Fig. 533, *II*), which limits the seat depth. The spring fails to provide reliable centring of the ball. The latter is difficult to keep from falling out during disassembly.

Cylindrical lock pins, or spring plungers with a spherical point (Fig. 534, *I*, *II*), are free from the disadvantages of spring-loaded

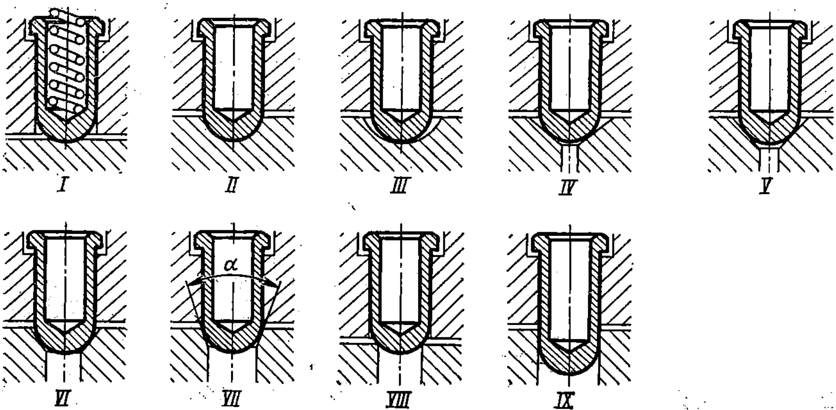


Fig. 535. Forms of seats for plungers with spherical points

balls. Such plungers are easy to prevent from falling out, e.g. by mounting them as is shown in Fig. 534, *II*.

The plunger of Fig. 535, *I* slides over a flat surface. Such an arrangement is used for stepless locking. The plunger here serves as a brake; the movable part is held in place by friction of the plunger against the flat surface.

In the assembly shown in Fig. 535, *II*, the plunger seat is spherical, which is disadvantageous in many respects. First, a spherical seat is difficult to machine; second, the force of locking is variable, for it depends on the situation of the point at which the plunger tip and the seat contact each other, i.e. on the geometrical accuracy of the contacting elements. Where the diameter of the seat sphere is greater than that of the plunger (Fig. 535, *III*), positive locking cannot be ensured.

A conical seat for plungers (Fig. 535, *IV-VII*) is better. By changing the cone angle, it is possible to adjust the locking force, i.e. the force whereby a plunger with the fully sunk tip retains the movable part in position.

The force necessary to dislodge the movable part fixed by a locking device of this type is determined from the relationship $T \approx \approx Q/\tan(\alpha/2)$, where Q is the pressure of the spring actuating the

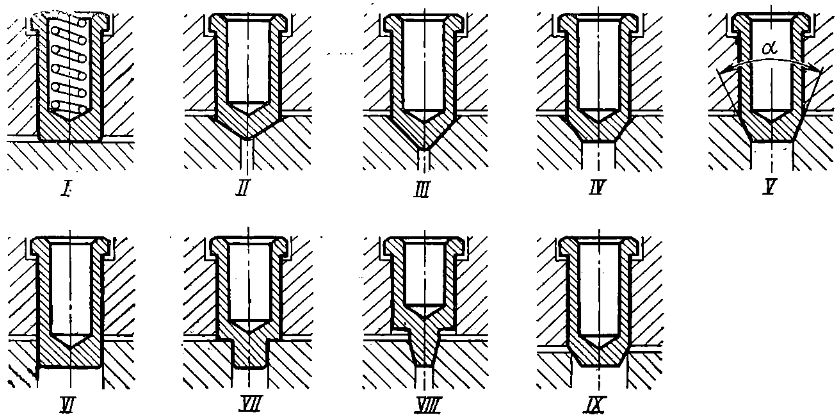


Fig. 536. Forms of points for cylindrical plungers

plunger, and α is the included angle of the conical seat (Fig. 535, *VII*). By reducing the cone angle to a definite value it is possible to obtain a self-holding effect and, therefore, positive locking.

Positive locking is also obtained where the tip or the cylindrical portion of the plunger enters a cylindrical bore (Fig. 535, *VIII*, *IX*).

Fig. 536, *I-IX* shows cylindrical spring plungers with points of various shapes. Conical points provide for greater accuracy of location than cylindrical or spherical points.

The effort to shift the movable part that carries a locking device along the stationary part develops a force on the conical surface of the plunger which tends to displace the plunger upwards (Fig. 537). This force

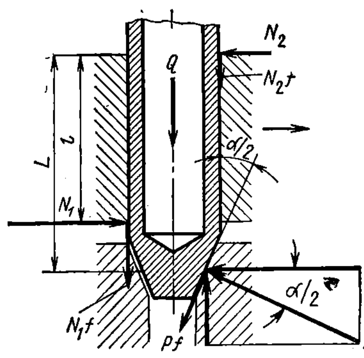


Fig. 537. Forces acting on conical-point plunger to displace it upwards

$$P = Q/\sin\frac{\alpha}{2} \tag{53}$$

where Q = pressure of the spring that loads the plunger, and $\alpha/2$ = = half the cone included angle.

Force P gives rise to reaction forces N_1 and N_2 at the extreme points of the plunger guide:

$$N_1 = P \cos \alpha/2 \frac{L}{l} \tag{54}$$

$$N_2 = P \cos \alpha/2 \left(\frac{L}{l} - 1\right) \tag{55}$$

The upward displacement of the plunger is resisted by friction forces $N_1 f$ and $N_2 f$ (where f is the coefficient of friction) and also by the axial component of friction force Pf that arises at the point of application of force P and equals $Pf \cos \alpha/2$.

The plunger is in equilibrium if

$$N_1 f + N_2 f + Pf \cos \alpha/2 = P \sin \alpha/2$$

Substituting expressions (54) and (55) for N_1 and N_2 into this equation, we obtain

$$\tan \alpha/2 = 2f \frac{L}{l} \tag{56}$$

The expression determines the marginal value of angle α at which the plunger still can be forced out of the seat. With smaller values of α self-holding takes place.

For locking devices where the plunger extends little from the guide, ratio L/l usually equals 1.2 to 1.3. The coefficient of friction can be assumed to equal 0.1.

Substituting these values into expression (56), we obtain $\tan \alpha/2 = 0.24$ to 0.26, wherefrom $\alpha/2 \approx 15°$ and the included angle $\alpha \approx 30°$.

The foregoing formulas do not take account of reaction forces due to friction in the guide of a movable part which carries the plunger. If this part rotates, such a reaction force is the force of friction in the pivot, which equals $fP \cos \alpha/2$. This force creates on the plunger a force that acts in the direction opposite to the rotation and is equal to $fP \cos \alpha/2\ (r/R)$, where r is the pivot radius, and R is the distance from the plunger to the pivot.

If the part with the plunger has a straight movement, the reaction forces are those opposing the movement; their magnitude depends on the construction and location of the guide ways. Owing to these additional forces, the self-holding effect practically takes place with $\alpha = 35$ to $40°$.

However, the coefficient of friction may vary, and therefore $\alpha < 25°$ should be adopted for reliable self-holding. If, on the contrary, disengagement of the locking device under the action of external forces is required, $\alpha > 60°$ should be specified. The same values of α hold true for devices with spherical-point plungers, where α is the included angle of the conical seat for the spherical point.

Locking devices in various forms are presented in Fig. 538. Among ball-type devices (Fig. 538, *I-V*), the one shown in Fig. 538, *II* features the adjustment of pressure of the loading spring.

The ball is kept from falling out by upsetting the edges of its hole made either directly in a part of a plastic metal (Fig. 538, *III*) or in an inserted bushing made of a plastic metal (Fig. 538, *IV*, *V*).

The locking devices of Fig. 538, *IV*, *V* are designed as self-contained units installed into an associated part.

Fig. 538, *VI-XIII* shows locking devices featuring cylindrical lock pins and plungers with spherical points. The devices of Fig. 538, *VII-IX* are arranged as separate units. In the design shown in Fig. 538, *IX* the plunger is kept from falling out by a cylindrical pin which extends through the slots in the housing and the plunger body.

Cylindrical lock pins providing positive locking are shown in Fig. 538, *X-XIV*. The lock pin and the hole must be chamfered to facilitate engagement. As with any device for positive locking, provision should be made for means whereby the lock pin is withdrawn from its hole.

Fig. 538, *XV-XVII* shows cylindrical lock pins with conical points; the one in Fig. 538, *XVII* is a part of a separate unit.

A lock pin with a prismatic point (Fig. 538, *XVIII*) which enter. a V-shaped seat on the mating part should be fixed against rotations This is achieved by the aid of flats provided on the lock-pin shank which extends through a correspondingly shaped hole in the housing.

The locking of bushings on shafts is illustrated in Fig. 539. Examples of elastic and positive locking are shown in Fig. 539, *I* and *II* and in Fig. 539, *III* and *VI*, respectively. In the latter case, provision is made for sinking the lock pins to allow disassembly of the joint.

In the assemblies shown in Fig. 539, *I-IV*, the lock pins get in a circular groove on the bushing and lock it axially while allowing its rotation on the shaft, whereas in those of Fig. 539, *V* and *VI* the lock pins enter mating bores in the bushing and so lock it against rotation and axial displacement.

In designs similar to those shown in Fig. 539, *IV* and *V*, the lock-pin shoulders should be shaped as spheres whose diameter is equal to that of the shaft bore for snug contact between these surfaces.

Concentric cylindrical parts are often fixed against relative axial displacement by spring rings. Such a ring is placed into a groove in the external part (Fig. 540, *I*); as the parts are assembled, the ring snaps into a circular groove on the shaft. The reverse arrangement is possible, i.e. a ring is placed into the groove on the shaft (Fig. 540, *II*) and, during assembly, gets into the groove in the external part.

To ensure reliable locking, it is necessary that in the first case the inner diameter d_1 of the spring ring in its free condition (Fig.

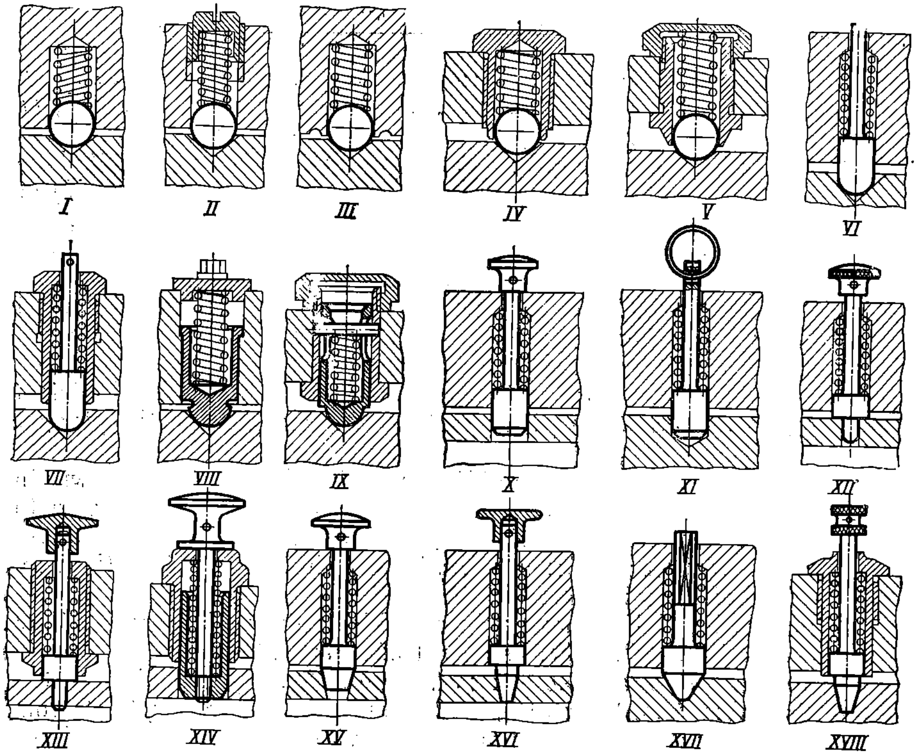


Fig. 538. Locking devices

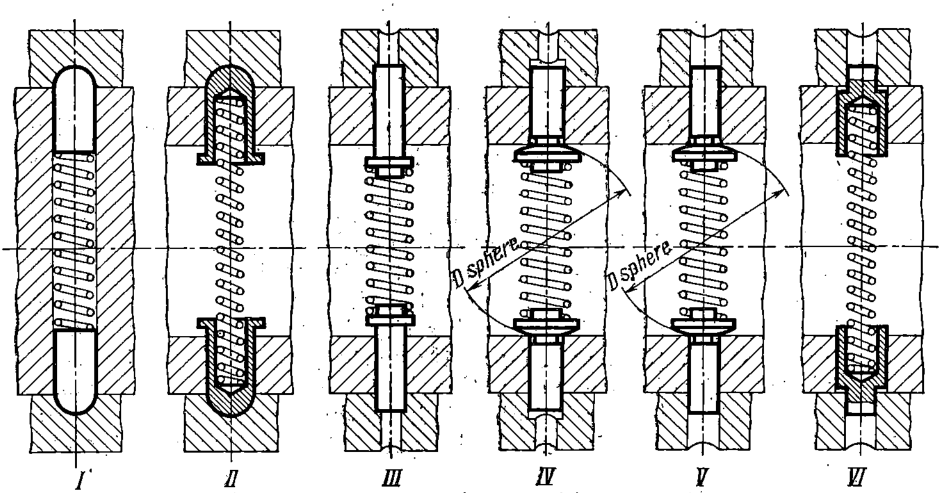
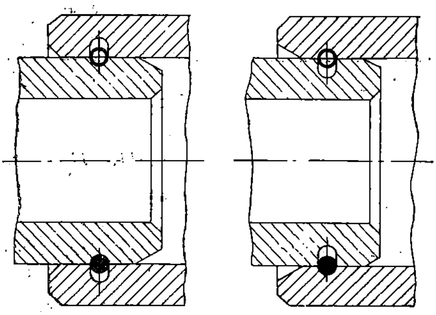


Fig. 539. Locking bushings on shafts

541, *I*) should be somewhat smaller than diameter d_2 of the shaft groove. In its working position, the ring should partly remain in the groove of the external part (amount *a* in Fig. 541, *III*).

In the second case, the outer diameter D_1 of the spring ring in its free state (Fig. 542, *I*) should be greater than diameter D_2 of the groove in the external part. In the working position, the ring should partly remain in the shaft groove (amount *a* in Fig. 542, *III*).

Spring rings of round section provide elastic locking. Where positive locking is required, use is made of spring rings of rectangular section (Fig. 543, *I*, *II* and *III*).

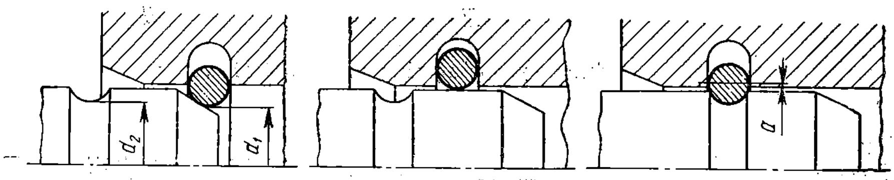


I *II*

Fig. 540. Locking concentric parts by spring rings. The ring is placed in hub groove (*I*) or in shaft groove (*II*)

With spring rings whose locking edge is defined by two conical flanks (Fig. 544, *I*, *II* and *III*), either elastic or positive locking can be obtained, depending on the included angle of the cones.

Fig. 545 shows typical constructions of clasp handles. In the handle presented in Fig. 545, *I*, lock pin *a* sliding in sleeve *b*, which is



I *II* *III*

Fig. 541. Locking by spring ring placed in external-part groove
I, *II*, *III*—assembly sequence

carried by hand lever *c*, enters a conical hole in stationary dial *d*. The lock pin is withdrawn from the seat by pulling handle *e*, and then it can be inserted into another hole in the dial.

The device of Fig. 545, *II* wherein the lock pin is connected with handle *e* by means of a multi-start thread provides greater convenience of operation. Here, the lock pin is retracted from the hole by turning the handle.

A handle which provides stepless locking is illustrated in Fig. 545, *III*. Here, the lock pin moves along a circular slot on the dial, whose centre coincides with that of the handle's rotation. Locking

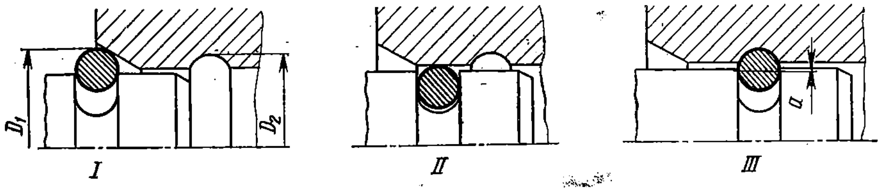


Fig. 542. Locking by spring ring placed in shaft groove
I, II, III—assembly sequence

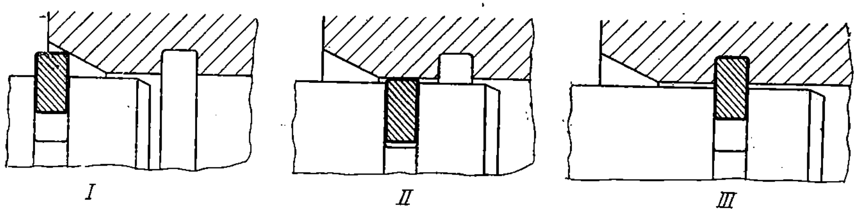


Fig. 543. Positive locking by spring ring of rectangular cross section

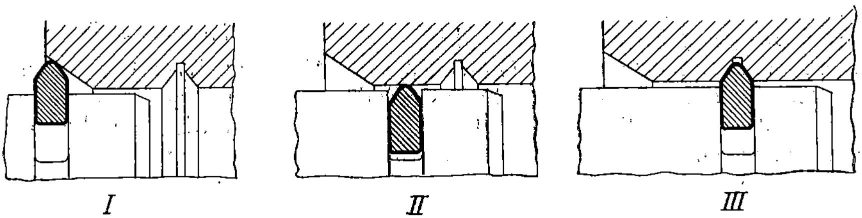


Fig. 544. Locking by spring ring with conical flanks

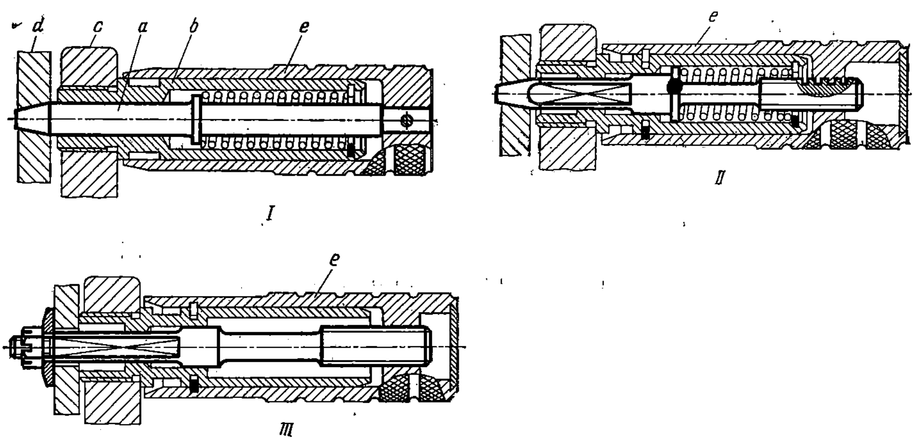


Fig 545 Clasp handles

in any position is effected by turning handle *e* round its axis and thereby drawing up the lock pin against the dial. The latter is released as the handle is turned in the opposite direction.

7.3. Axially Loaded Rotary Joints

Such joints provide free rotation of one part relative to another and locking against relative axial displacement. An example is a joint wherein the closing element of a valve is connected to a screw rod which causes the closing element to open and close the valve.

Fig. 546 shows an assembly where the rod is provided with a head whose diameter is greater than that of the rod shank, and where the locking element can be mounted from the other end of the rod.

In the assemblies presented in Fig. 547, the locking element can be mounted from any side. Fig. 547, *I* shows a rod fixed by a set screw which enters a circular groove on the rod end. A drawback to this design is eccentrical application of load as the rod travels upwards, so that only light loads can be carried during the movement in this direction. The use of two or three symmetrically located set screws does not help much since in any case only one of the screws will bear the load due to inevitable deviations from the true position of the screws along the rod axis.

A rod can be fixed by a pin (Fig. 547, *II*) passed through its hole into a circular groove in the mating part. The pin is kept from falling out by a threaded plug.

Fixing by flange-type elements is shown in Fig. 547, *III* and *IV*. The flange in Fig. 547, *III* fits into a circular recess on the rod; that in Fig. 547, *IV* comprises two parts with a spigot feature to fix them against an angular shift around the bolts.

In the assemblies in Fig. 547, *V* and *VI*, the rod is fixed by half-rings secured with a nut (Fig. 547, *V*) or with a retaining ring (Fig. 547, *VI*). In the assemblies of Fig. 547, *VII-IX*, locking is effected by retaining rings.

A retaining ring is placed into a circular groove in the external part (Fig. 547, *VII*) so that it snaps into a groove on the rod as this is pushed in. To release the rod for disassembly, the ring is expanded with tongs reaching it through the recess in the external part.

Another variation where the retaining ring is placed in the groove of the external part and snaps into the groove on the rod during assembly is shown in Fig. 547, *VIII*. Disassembly, however, is effected by contracting the ring through radial holes in the external part.

In an assembly of Fig. 547, *IX*, the retaining ring is locked in position by a nut.

The rod can also be fixed with a soft-steel wire passed through circular grooves of half-round section made on it and in the mating

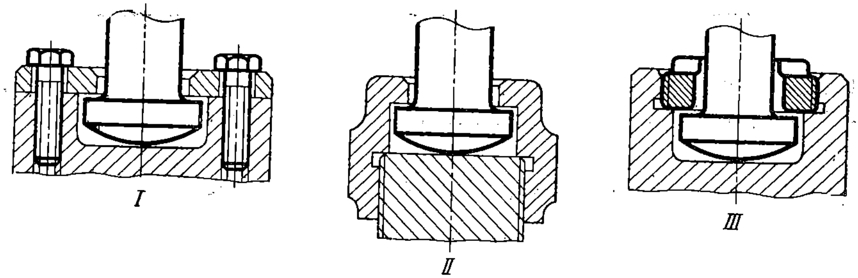


Fig. 546. Axially loaded rotary joints. Locking element is installed from the opposite end of rod. Rod head is fixed by flange (*I*) and by nuts (*II*, *III*)

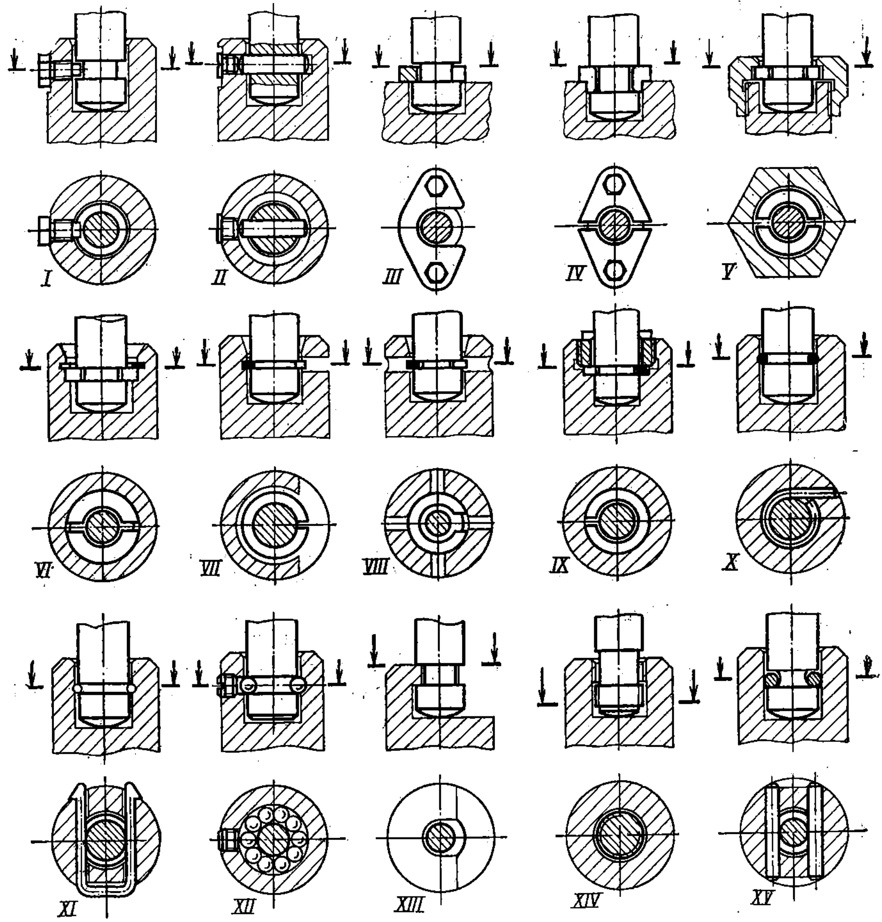


Fig. 547. Rotary joints in various forms

part (Fig. 547, *X*). In another version (Fig. 547, *XI*), a clip made of soft-steel wire is used to fix the rod in much the same manner.

Fixing with balls placed through a lateral hole into circular grooves of half-round section made in the outer part and on the rod is shown in Fig. 547, *XII*. Unlike most of the assemblies just described, this joint is capable of transmitting great forces in both directions with minimum resistance to rotation. Here, the rod end face

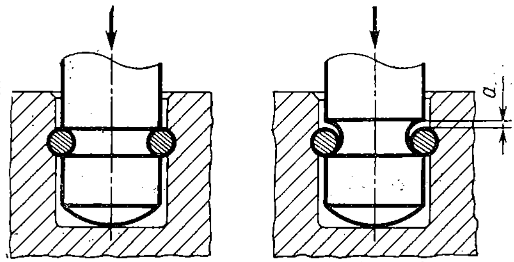


Fig. 548. Securing with pins
I—incorrect; *II*—correct

need not bear against the mating part. The joint requires increased accuracy of manufacture. The grooves for balls must be hardened to Rc 45.

A widely used method is mounting a rod into its mating part through the slot in the overhanging portion, as is shown in Fig. 547, *XIII*. Here, the rod must be fixed against lateral displacement relative to the mating part.

Fig. 547, *XIV* illustrates a simple method whereby a threaded end of the rod is screwed in through a threaded hole in the mating part. The joint is used where upward-stroke loads are small. If the rod is subject to rotation during the upward movement, the thread should have a hand opposite to the direction of rotation.

Fixing with two pins inserted through holes in the outer part into a circular groove of half-round section made on the rod has found extensive application (Fig. 547, *XV*). The holes in the outer part are usually drilled with the use of a jig prior to machining the central bore, which helps to provide accurate location of the pins necessary for proper functioning of the joint. To ensure that the rod end face bears against the bottom of the bore in the mating part, the radius of the rod's groove section is made somewhat greater than that of the pins (clearance *a* in Fig. 548, *II*). The centres of the pin holes are so disposed that they coincide with the rod generators or, better still, lie slightly closer to the rod axis.

A permanent joint obtained by squeezing the outer part around the rod on a rotary die press so that the part material fills the rod grooves is shown in Fig. 549. To ensure free rotation of the rod, it is

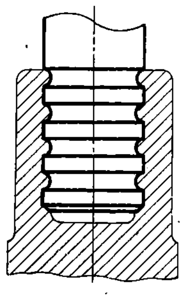


Fig. 549. Joining by swaging

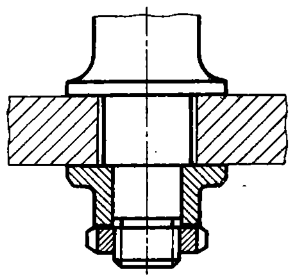


Fig. 550. Securing with nut and spacer

coated, prior to the squeezing operation, with colloidal graphite or some other grease with similar properties.

Where the rod end is accessible (Fig. 550), the rod is fixed by a nut through a spacer.

Coaxial cylindrical parts which should freely rotate on each other are fixed against relative axial displacement with split spring rings.

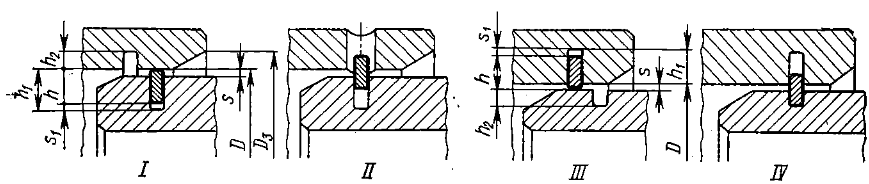


Fig. 551. Locking by spring rings

Such a ring is placed into a groove in the inner part (Fig. 551, *I*). The latter is then inserted into the outer part, and the ring gets into its groove (Fig. 551, *II*).

Depth h_1 of the groove on the inner part should be adequate to allow the ring to be fully sunk in it during assembly, i.e. $h_1 \geqslant h - s + s_1$, where h is ring height, s is radial clearance between the mating parts, and $s_1 \approx s$ is additional clearance to allow for dimensional variations.

Depth h_2 of the groove in the outer part is made equal to about 0.5 h. The outside diameter of the spring ring in its free condition equals $D_2 = D + h_2$, where D is the bore diameter. To facilitate assembly, the bore is provided with a slow-taper chamfer with diameter D_3 slightly larger than D_2.

The assembly may be made separable by providing radial holes in the walls of the outer part (Fig. 551, *II*), which make it possible to contract the ring.

Fig. 551, *III* and *IV* shows an assembly where the spring ring is placed in the groove of the outer part. As in the foregoing, the groove depth is $h_1 = h - s + s_1$. The depth of the groove in the inner part is $h_2 \approx 0.5\, h$. The internal diameter of the ring in its free condition is $D_2 = D - h_2$. This assembly is practically inseparable and therefore is seldom used.

7.4. Spherical Joints

These are exemplified by self-aligning bearings, assemblies which operate in misalignment conditions, ball-and-socket assemblies of rods, links, etc. The main methods for arranging spherical joints, illustrated in Fig. 552, are as follows.

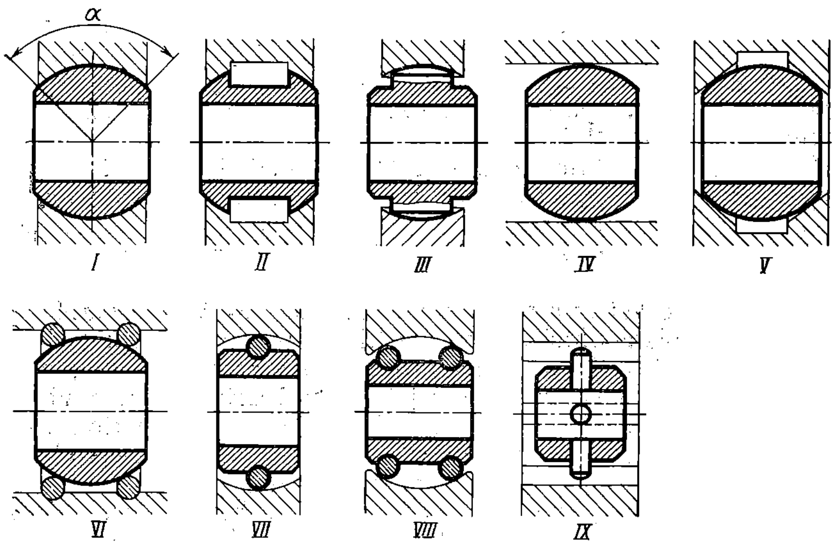


Fig. 552. Arrangement of spherical joints

Sphere in a Spherical Socket (Fig. 552, *I*). This is the most commonly used joint capable of carrying heavy radial and axial loads (with a great angle of contact α of its bearing surfaces). Variations of this joint are shown in Fig. 552, *II*, where the ball and socket are recessed in the middle (an arrangement used for predominantly axial loads) and in Fig. 552, *III*, where the ball is relieved on the ends (a joint used for small, mainly radial, loads).

Sphere in a Cylindrical Socket (Fig. 552, *IV*). The joint is simpler than the preceding one as regards its producibility. Unlike the preceding assembly, here the bearing surfaces contact each other along a circle, and hence the joint can carry only small loads.

Sphere in a Conical Socket (Fig. 552, *V*). The joint can carry smaller loads than the first arrangement. The clearance between the bearing surfaces is adjusted by displacing the cones closer to each other or farther apart.

Sphere in a Socket Formed by Rings (Fig. 552, *VI*). The joint is basically similar to the preceding one. The rings may be split, like retaining rings, for easier assembly of the joint.

Ring in a Spherical Socket (Fig. 552, *VII*). This joint is designed for small loads.

Two Rings in a Spherical Socket (Fig. 552, *VIII*). The joint is designed for small radial and axial loads.

Shown in Fig. 552, *IX* is a joint kinematically equivalent to the spherical joint. Rotation in any direction round the central point is provided by crosswise-arranged cylindrical pins which engage in associated slots in the socket. This joint can carry small radial loads.

Construction of Ball Joints. Ball-and-socket joints are used to transmit forces to pulling or pushing levers or other elements in three-dimensional link mechanisms.

The most typical field of application of such joints is lever-type control mechanisms, although they may sometimes be found in lever-type power transmission where working loads are not too high.

Fig. 553 presents ball joints used in engineering practice. The simplest joint (Fig. 553, *I*) consists of a pivot with a ball point which fits into the socket in a connecting rod. The socket is separable and has a parting plane which coincides with the rod axis. The detachable part is fastened to the rod head with screws through a gasket, which is used to adjust clearance in the joint. The bearing surfaces subjected to friction must be hardened, which is done by induction hardening, cyaniding, and other methods. These surfaces must also be lubricated, which is done mainly by greasing.

In the joint of Fig. 553, *II*, the spherical socket is formed by two hardened inserts fixed in the rod head by a lid which is held in place by curling the edge of the bore. This is a permanent joint.

Ball joints designed for simpler manufacture of the sockets have the ball mounted between conical surfaces (Fig. 553, *III*), between retaining rings (Fig. 553, *IV*) and between a single retaining ring and the bottom of the socket (Fig. 553, *V*). These joints are used for smaller loads than those with spherical sockets.

A reverse arrangement is shown in Fig. 553, *VI*; here, the connecting rod is provided with a ball point, and the pivot, with a socket.

Some ball joints can be adjusted for tightness (see Fig. 553, *VII*-*IX*). In the joint of Fig. 553, *VII*, the adjustment is effected by

tightening screw cap *a* with a spherical socket on its end, whereupon the ball point is driven deeper into spherical insert *b*. In another version (Fig. 553, *VIII*), the ball point fits into two spherical inserts

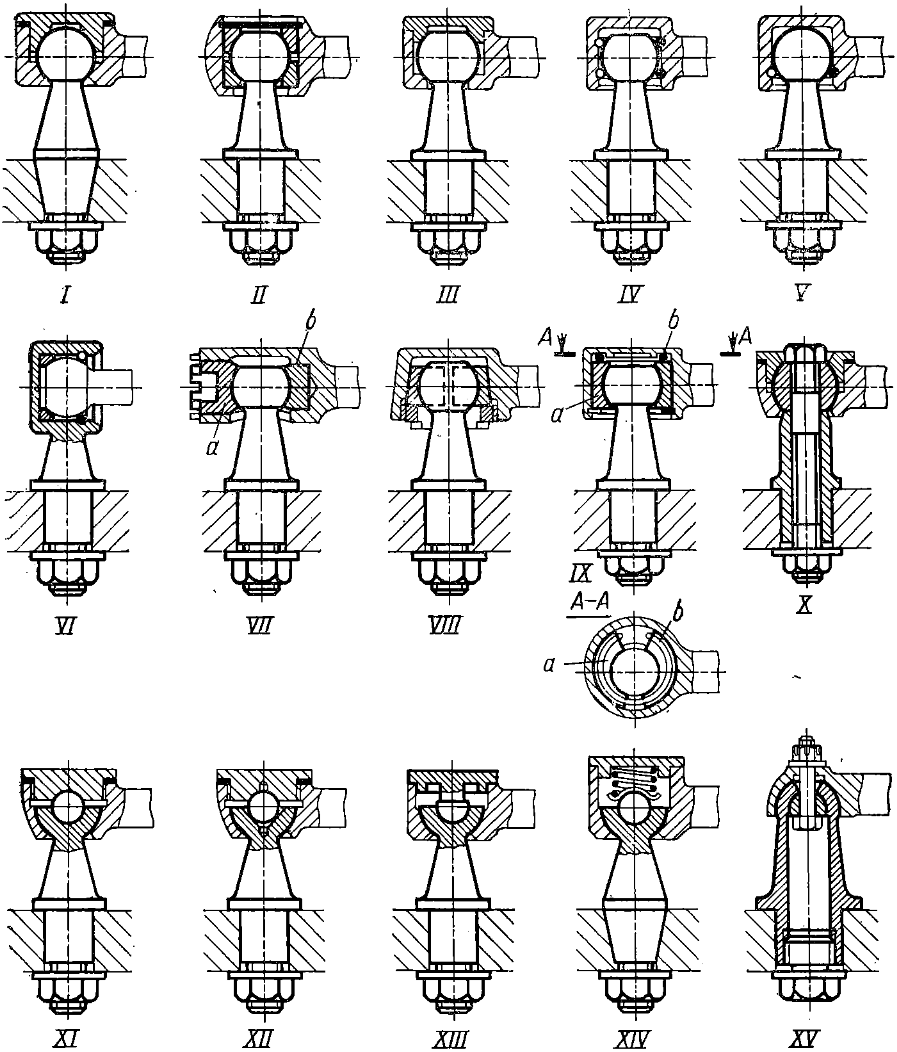


Fig. 553. Spherical joints in various forms

with tapered outer surfaces, which are placed into a tapered bore in the rod head. The adjustment is done by tightening a nut, which acts on the inserts.

Fig. 553, *IX* illustrates self-adjustment for tightness. The ball point is enclosed by spherical inserts *a* so shaped that the ball centre is disposed eccentrically with respect to their outer surfaces. Spring ring *b* urges the inserts to move into the wedge-like space between the ball point and the walls of the rod head, thereby tightening the joint.

A pivot made up of several components for simpler machining of the ball element is shown in Fig. 553, *X*.

In the joints shown in Fig. 553, *XI—XIII*, the pivot head is shaped as an external half-sphere with a spherical socket which houses a small-diameter ball. The joint is tightened by means of a threaded cap.

The joint of Fig. 553, *XIV*, tightened by a spring, is designed for small loads.

A compact assembly is shown in Fig. 553, *XV*. Here, a hollow pivot is provided with a spherical head which has a socket inside to receive a spherical insert. The joint is tightened with a bolt and nut.

7.5. Spherical-Point Rods

These are used in various linkage mechanisms to transmit pushing forces. One example is a follower of the camshaft in some internal combustion engines.

Fig. 554 presents typical solid rods with spherical points and heads. The rods in Fig. 554, *I-V* are made from bar stock; those in

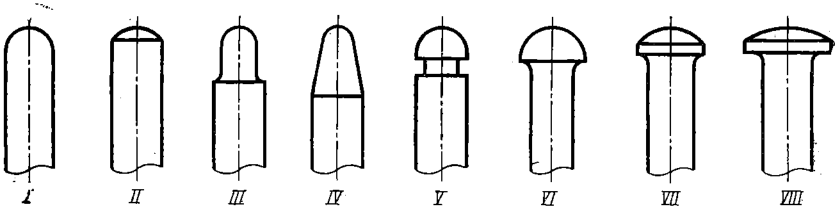


Fig. 554. Spherical points of push rods

Fig. 554, *VI-VIII* require heading operations. The working surfaces are hardened to *Rc* 58-62 by various heat-treatment methods.

Push rods in mechanisms of high-speed machines move with substantial accelerations. To reduce the inertial forces, use is made of light-weight rods comprising a tubular body and a spherical point fixed therein. Such rods in various forms are shown in Fig. 555, *I-XVI*.

The joint of a tube with a point is subjected to cyclic loads which are similar in character to impact loads. For this reason, special emphasis is placed on the strength and rigidity of the joint.

The point is inserted into a reamed tube by a press fit (Fig. 555, *I*). To prevent expansion of the tube, its end is wedged into a recess

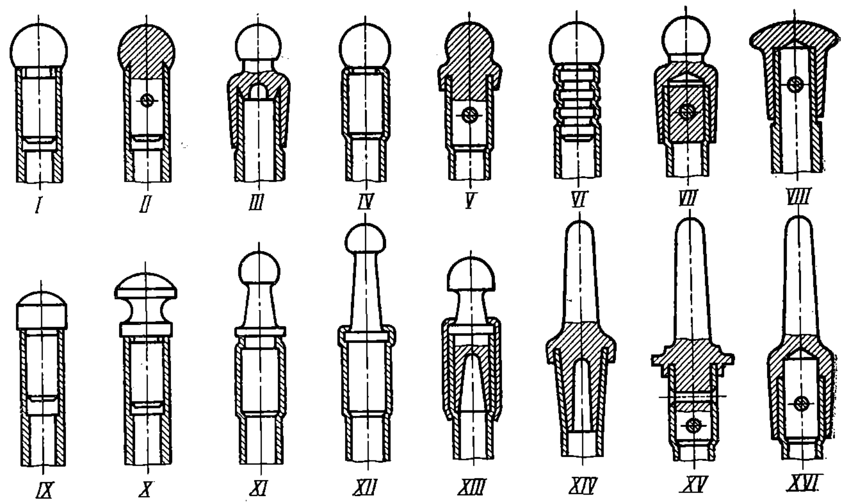


Fig. 555. Joining spherical points to tubular rods

on the point underside (Fig. 555, *II*). The joint is additionally secured with fastening pins or rivets (in this case, the point's shank

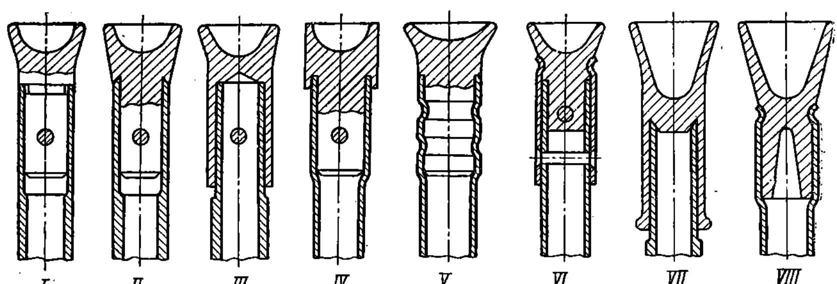


Fig. 556. Joining spherical sockets to tubular rods

must be unhardened). Sometimes, the point is mounted on a turned external surface of the tube (Fig. 555, *III*). These methods tend to weaken the tube.

A better method is to insert the point into a preformed tube (Fig. 555, *IV*, *V*, *XI*, *XII*) and then to swage the latter on a rotary die press. Sometimes, the tube material is forced into grooves on the point's shank (Fig. 555, *VI*).

The joint can be reinforced by introducing inserts (Fig. 555, *VII*) or external collars (Fig. 555, *XIII*).

A strong joint is shown in Fig. 555, *XIV*. The point has a taper shank, so that when pressed into a cylindrical tube, it expands the tube end. The flange of the point is then rolled back against the tube.

In some application the rod carries a spherical socket which accommodates the spherical points of driven parts.

Some varieties of push rods with spherical sockets and methods for their fixing in tubular shanks are presented in Fig. 556, *I-VIII*.

7.6. Gear-Rim Assemblies

Fig. 557 illustrates methods for assembling bronze rims with cast iron and steel discs. The assembly is effected mainly by forced

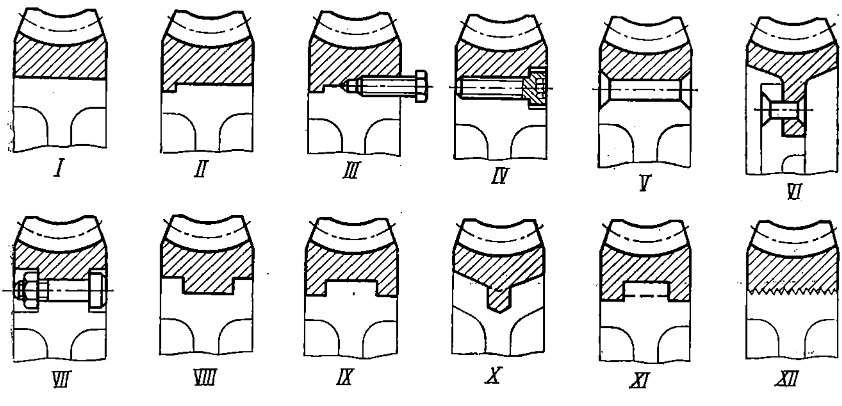


Fig. 557. Assembling bronze gear rims to cast iron or steel discs

fits. Some rims are mounted so that their end faces are flush with the end faces of the discs (Fig. 557, *I*), others are provided with shoulders for axial location on the discs (Fig. 557, *II*).

The amount of interference provided by forced fits (Fig. 557, *I*) is fairly adequate for the transmission of torque and for the axial locking of the rim on the disc, especially if the former is large in diameter. Nevertheless, rims are additionally secured by screws cut off flush with the rim end face after assembly (Fig. 557, *III*) or sunk into counterbores (Fig. 557, *IV*), and also by rivets (Fig. 557, *V*).

Gear hobbing and finish facing operations are carried out after assembly of the rim with the disc.

A rim riveted to a disc is shown in Fig. 557, *VI*, and bolted, in Fig. 557, *VII*, the latter assembly being separable.

In some cases, bronze rims are moulded on the discs. The adhesion of the rim to the disc is improved by introducing steps (Fig. 557, *VIII* and *IX*) or tenon-like features (Fig. 557, *X* and *XI*) on the disc. The simplest method, which is fairly reliable, is to groove the mating surface on the disc prior to moulding (Fig. 557, *XII*). Rim moulding, however, is not used for critical applications. Here, it is more advisable to use rims in the form of heat-treated bronze forgings mounted on discs.

Chapter 8

Locking Screw Fasteners

8.1. Locking Methods

All screw fasteners in machines must be reliably secured against loosening. Nonobservance of this requirement is fraught with extremely grave consequences. Nuts or bolts breaking loose inside a mechanism have been known to cause heavy damage and breakdown of costly equipment.

Two principal methods of locking screw fasteners are distinguished. **Positive locking** means that the fastener to be fixed is connected with the locking element by a rigid link, so that the locked fastener cannot be loosened without breaking or deforming that link. Locking by this method can be exemplified by the use of split cotter pins, safety washers, locking plates, wire binding, etc.

The other method involves the increase of friction between the locking and locked elements and is therefore called **friction locking**. Here, use is made of lock nuts, spring washers, self-locking nuts, etc.

Friction locking is less reliable than positive locking, for there is always the danger that the forces of friction may decrease and the screw joint may consequently slacken. For this reason, all critical threaded assemblies and also those placed inside machines are fixed only by positive-locking means (mainly split cotter pins). Screw joints whose loosening will not cause breakdown to the machine and also externally located fasteners which are accessible for inspection may be fixed by friction locking. In this case, however, the fasteners must be periodically inspected and tightened.

A variety of friction locking which may be called **elastic locking** involves the introduction of an elastic element to maintain constant tension in the screw joint. The greater the elasticity of the joint, the more reliable is its friction locking. Here, the force of friction between the locked and the locking component will be maintained under service conditions which involve residual strain within the joint, vibrations, pulsating loads, etc. Lock nuts hardly give any elasticity to the joint, spring washers are only marginally effective, but there are some elastic-locking devices which provide very high elasticity indeed.

Some locking methods combine the principles of positive locking and friction locking. One instance is the use of ratchet-type washers. Here, the locking effect is achieved partly through increased fric-

tion in the screw joint due to elastic deformation of the washer in the process of tightening, and partly through positive connection between the nut and the part underneath, obtained as the washer teeth bite into the bearing surface of the nut and that of the part.

The nut can be fixed on the bolt and on the base (that is, the part underneath). More specifically, the arrangements used are as follows.

(1) The **nut is locked on the bolt thread.** Here, friction between the nut and bolt threads is increased by various methods to tighten the joint. The locking is effected by means of interference-fit threads, lock nuts, spring washers, self-locking nuts, etc.

(2) The **nut is attached to the bolt shank.** To effect this arrangement, use is made of various locking elements which enter holes or recesses on the nut and the bolt shank, e.g. split cotter pins, tongued washers, etc. This locking is positive.

(3) The **nut is attached to the base.** Here, rigid or elastic connection is provided between the nut and the base or some elements thereon. The locking is effected by tab washers, screwed-down washers, wire bindings, etc. However, the bolt must be fixed against rotation relative to the base lest it should screw out of the nut.

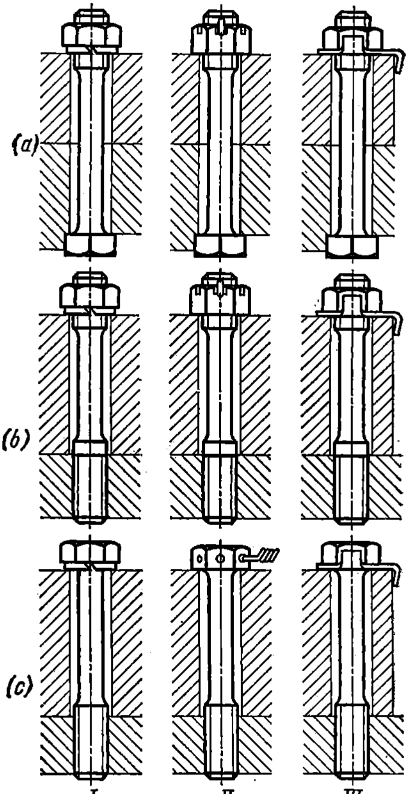


Fig. 558. Methods for locking fasteners

Fig. 558 illustrates various methods for locking fasteners. Friction locking is here exemplified by the use of split spring washers, and positive locking, by the use of split cotter pins, tab washers, and wiring. Bolts (*a*) and studs (*b*) can be fixed by all available methods: friction locking, positive locking with the nut attached to the shank of the bolt or stud, and positive locking with the nut attached to the base. Gap screws (*c*) can be fixed by friction locking (*I*) and by positive locking of the screw on the base (*II*, *III*).

Irrespective of the locking method, the elasticity of bolts should be increased. Locking becomes more reliable owing to the maintained tension in the joint.

In service, assemblies with short rigid bolts (Fig. 559, *I*) slacken quickly, because residual strain inevitably developing with time in the thread and on the bearing surfaces is commensurable in magnitude with the bolt elongation caused by the initial tightening. A constant tension in such joints is impossible to maintain, especially under service conditions involving jarring, vibration, and pulsation of the working load. Positive locking (see Fig. 559, *II*) provides

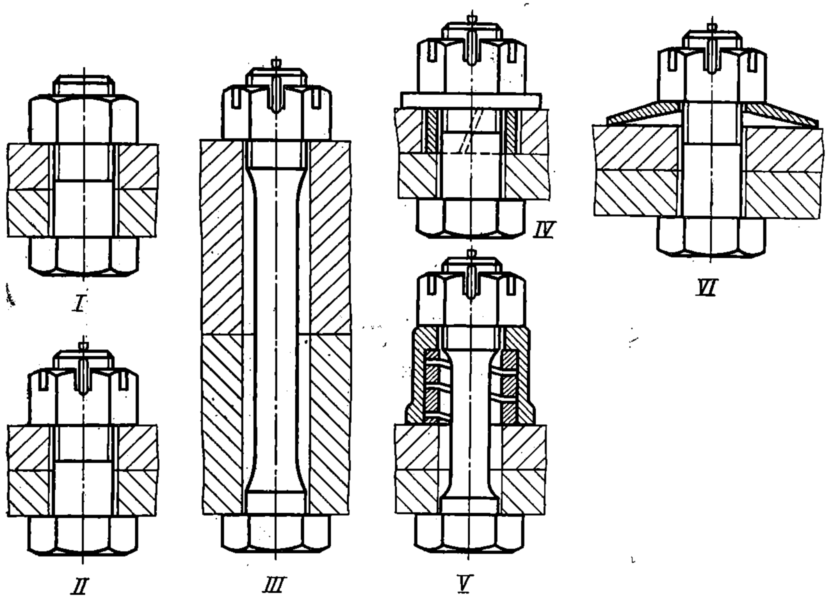


Fig. 559. Rigid and elastic bolts. Methods for increasing elasticity of threaded assemblies

no satisfactory solution here; the split cotter pin will only prevent the bolt and nut assembly from falling apart and not from loosening in time and becoming inoperable.

Where short rigid bolts have to be used (for fastening sheet-metal structural elements, panels, etc.), the fasteners must be periodically inspected and tightened, if need be (as is done, for instance, in servicing motor cars, where the nuts on the bolts fastening the chassis and body are regularly tightened). Split cotter pins for nut locking are not used here, since they would only make it difficult to tighten up the nuts.

A rational design solution is to increase elasticity of the screw joint. Slender elastic bolts (Fig. 559, *III*) can hold the nuts from loosening, and split cotter pins are used in such applications as a safeguard only. The use of long bolts, however, is often limited for

dimensional reasons, and then special elastic elements are resorted to (Fig. 559, *IV-VI*).

A similar effect can be obtained with greater elasticity of fastened components. Fig. 560, *I* and *II* shows the fastening of elastic flanges,

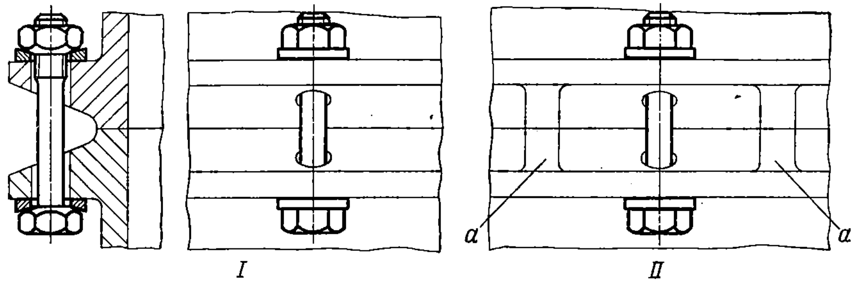


Fig. 560. Methods for bolting together elastic flanges (with flanges *II* elasticity is limited by ribs *a*)

and Fig. 561, *I*, *II* and *III*, that of an elastic cover. Needless to say, such designs can be adopted if the parts are made of strong materials with a high modulus of elasticity. To exclude overstrain in the course of tightening, use is made of special limiting features on the parts

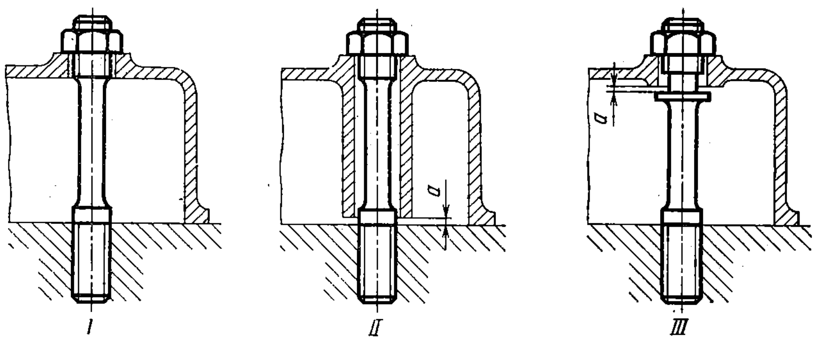


Fig. 561. Stud bolt used to fasten elastic covers

or fasteners (Fig. 561, *II* and *III*). The amount of strain is determined by clearance *a* taken up during tightening.

Nuts constantly loaded by powerful springs (Fig. 562, *I*) are fixed by the friction locking method. However, vibrations and dynamic loads necessitate additional locking of the nut by suitable means (Fig. 562, *II*).

Permanent locking is used where the nut is mounted on the bolt for the whole service life.

Fig. 563 illustrates different kinds of permanent locking: complete (*I*) or partial (*II*) welding or brazing of the nut to the bolt, notching (*III*) with a centre punch, upsetting (*IV*), staking (*V*, *VI*),

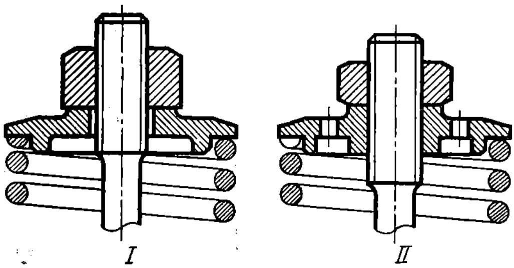


Fig. 562. Adjusting spring pressure. Nut is locked by spring pressure (*I*) or by lock nut (*II*)

squeezing of the nut collar on the bolt (*VII*), expanding of the bolt end with a taper pin (*VIII*), and fastening of the nut to the bolt with a pin (*IX*).

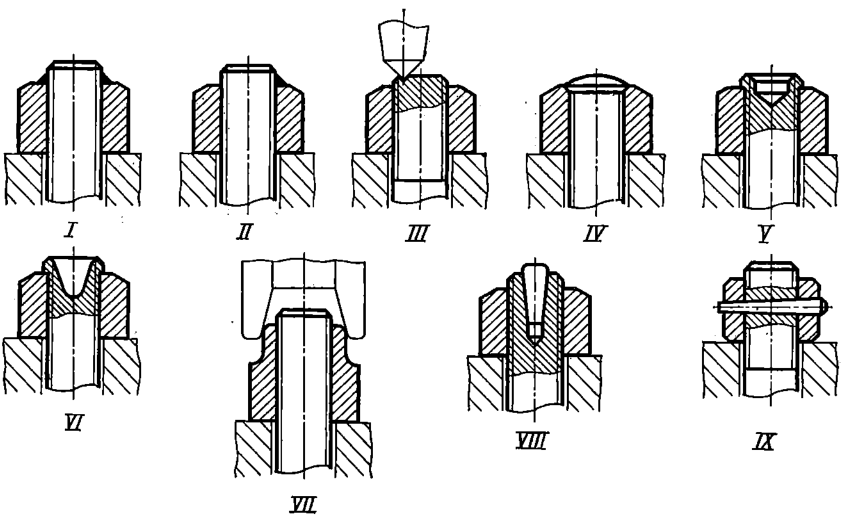


Fig. 563. Permanent locking of nuts on bolts

The simplest of these methods is welding, especially partial (one drop of liquid metal is quite enough).

8.2. Locking with Lock Nuts

Lock nuts (Fig. 564) are rarely used as they fail to provide reliable locking. Another drawback to lock nuts is that they cause elongation of the bolt during tightening and take the entire load while the main nut becomes unloaded (Fig. 565). That necessitates the use of the reverse arrangement, where the lock nut is placed under the main nut (Fig. 564, *II*), which provides favourable load distribution.

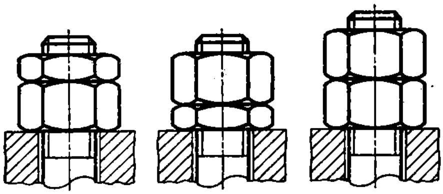


Fig. 564. Locking with lock nuts

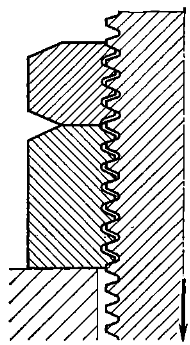


Fig. 565. Threads of the main nut and lock nut disposed relative to bolt threads after lock-nut tightening

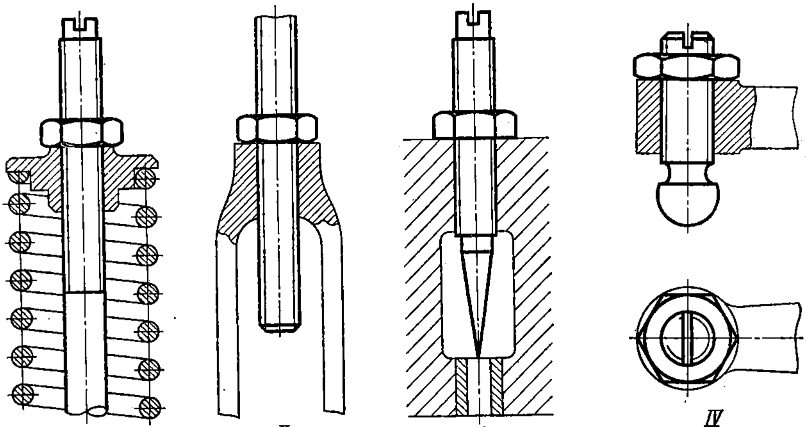
The lock nut is sometimes made equal in height to the main nut (Fig. 564, *III*).

Lock nuts are indispensable where the nut must be fixed on the bolt in any desired position, especially with a long travel of the nut along the bolt. Applications of this type are illustrated in Fig. 566; here, the lock nut, while being tightened on the threaded rod, bears against the part into which the rod is screwed.

Lock nuts are often used in assemblies for axial adjustment of antifriction bearings (Fig. 567). Special lock nuts are shown in Figs. 568 and 569.

Fig. 570 presents lock nuts with skirts tapered for better engagement with the main nuts. Such nuts can be split (Fig. 570, *II* and *III*), which increases compliance of the skirt in a radial direction. Excessive compliance may hamper full tightening of the lock nut due to its jamming at the final stage of tightening. The disadvantages of such nuts are that the tapered mating surfaces require a complicated manufacturing process and that they are subject to higher contact stresses.

Shown in Fig. 571 is an elastic nut called Palnut after the name of the manufacturer. The nut is made of hardened sheet metal. It has



I II III IV

Fig. 566. Lock nuts used in assemblies for spring-pressure adjustment (*I*); turnbuckle adjustment (*II*); needle valve adjustment (*III*); ball-end push rod adjustment (*IV*)

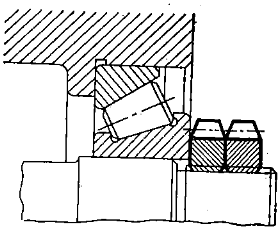


Fig. 567. Lock nut used to secure axial adjustment nut in taper-roller bearing assembly

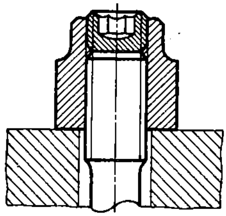


Fig. 568. Male threaded lock nut screwed into tapped hole

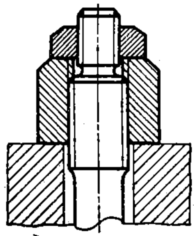
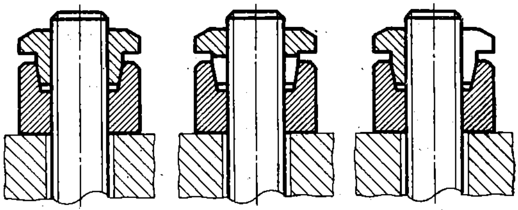


Fig. 569. Lock nut screwed on special bolt end (the bolt has a right-hand screw thread, and its end, a left-hand screw thread)



I II III

Fig. 570. Tapered lock nuts

I—solid; *II*—with radial slots; *III*—split

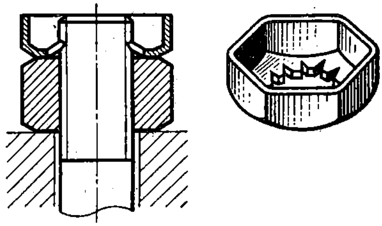


Fig. 571. Palnut elastic nut

a single turn of thread formed by internal teeth disposed on a helix. The nut is lightweight and elastic, which makes for its reliable engagement with the bolt. Recently, elastic lock nuts have been made integral with the main nuts.

8.3. Locking with Split Cotter Pins

This is a very reliable method of locking used in the most critical applications.

Fig. 572 illustrates an outdated method of locking with split cotter pins for comparison with modern methods. Its drawbacks are

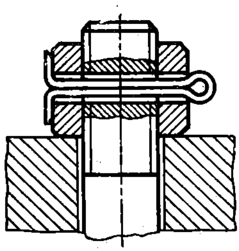


Fig. 572. Locking with split cotter pin. Outdated method

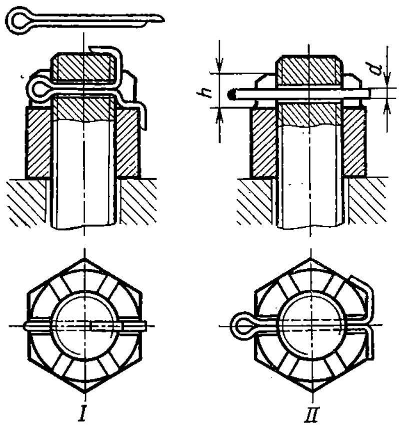


Fig. 573. Locking with split cotter pins by bending the pin ends against bolt end face and nut flat (*I*) and against nut flats (*II*)

that the nut is weakened in the most heavily loaded working region and its angular positioning range as permitted by the cotter pin is limited.

In modern designs (Fig. 573), the nut is allowed to move for tightening within a wider range owing to slots provided in its top portion (the number of slots being usually six for hexagon nuts). Depth h of the slots is substantially greater than diameter d of the cotter pin. The latter, made from wire of half-round section, is inserted into one of the slots and into a hole in the unloaded upper end of the bolt. The cotter pin ends are then spread.

Two methods of setting split cotter pins are in use. According to the first method (Fig. 573, *I*) the cotter pin is set so that its eye is in a plane parallel to the bolt axis. One end of the pin is bent against the bolt end face, and the other, against one of the nut flats. Accord-

ing to the second method (Fig. 573, *II*), the cotter pin eye is set in a plane square to the bolt axis, and the pin ends are spread against the nut flats.

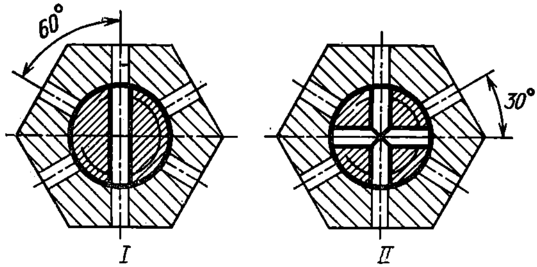


Fig. 574. Nut rotation angle for locking with cotter pins
I—with a single pin hole in bolt; *II*—with two mutually perpendicular holes

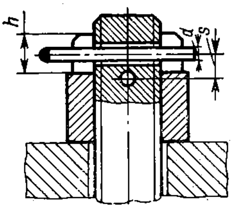


Fig. 575. Assembly with increased nut-locking range

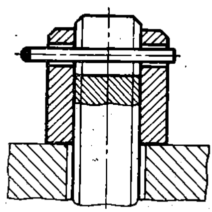


Fig. 576. Locking with cotter pin passed through holes in nut and slot in bolt

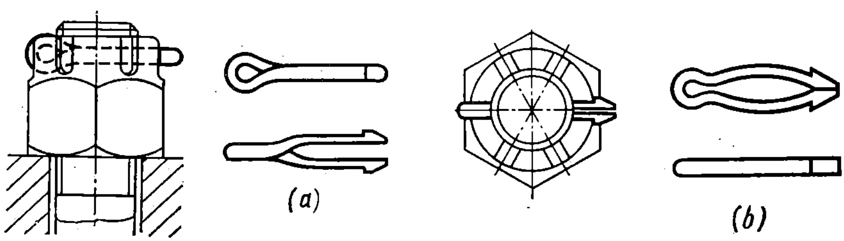


Fig. 577. Elastic cotter pins (*a*, *b*)

The first method is more frequently used for simpler mounting and smaller size of assembly. The second method, however, offers a wider positioning range wherein the tightened nut can be locked.

Nuts with six slots allow locking at every 60° of rotation. With a thread pitch of about 1.5 mm, this corresponds to a 0.25 mm elongation of the bolt, so that fine adjustment of the tightening load can hardly be achieved.

Sometimes the bolt is provided with two mutually perpendicular cross holes (Fig. 574) for finer adjustment. In this case, the nut can be locked at every 30° of rotation. The locking range of the nut along the bolt axis is $t=h-d$, where h is the slot depth and d is the cotter pin diameter. To increase this range, the holes for cotter pins are separated axially (Fig. 575) by amount $s = h - d$ (not more). This arrangement is used for long bolts whose elongation due to tightening can be considerable (0.5 mm and over.)

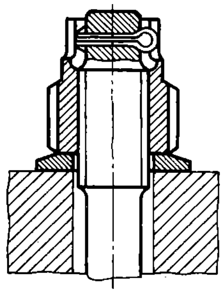


Fig. 578. Serrated nut locked with elastic cotter pin

Sometimes the slot for a cotter pin is made on the bolt and the hole, in the nut (Fig. 576).

Split cotter pins are made of soft steel; they are disposable elements replaced by new ones after each disassembly. Elastic cotter pins made from hardened steel for repeated use (Fig. 577) have also found application in engineering practice. Their prongs are provided with fangs that reliably secure the pin after its insertion.

An instance of locking a serrated nut with an elastic cotter pin extended through the bolt end is shown in Fig. 578.

8.4. Locking with Washers

8.4.1. Safety Washers

Locking with safety washers is a common method of positive locking. Safety washers are made of soft steel and provided with projections such as tabs or tongues. Fig. 579, *I* and *II* shows the most commonly used forms of safety washers known as tab washers.

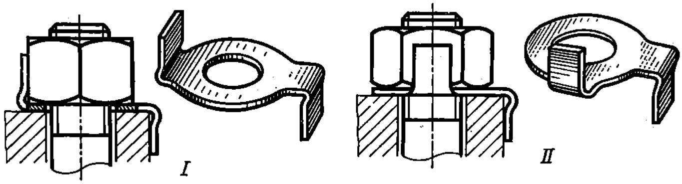


Fig. 579. Locking with tab washers

When a tab washer is set in place, one of its tabs is secured to the base (usually bent against the nearest edge of the base) and the other tongue is bent against a flat of the nut. In this way rigid connection is ensured between the base and the nut.

Tab washers can be installed by other methods, using any suitable feature of the base situated near the nut. Sometimes, special fea-

tures have to be provided for the purpose: a hole near the nut (Fig. 580, *I*) to receive the washer tab, or a pin extended through a

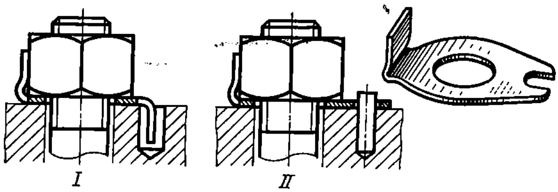


Fig. 580. Fixing tab washers to base

hole in the tab (Fig. 580, *II*). The tab can also be fixed by an adjacent screw.

Tab washers can be used to lock cap screws and nuts screwed on studs. In locking the nuts screwed on bolts (Fig. 581), the bolt heads should be fixed against rotation. Otherwise (Fig. 581, *I*), the bolt can unscrew from the nut. A proper arrangement is shown in Fig. 581, *II*, where both the nut and bolt are fixed. A tab washer should be firmly and rigidly fixed on the base.

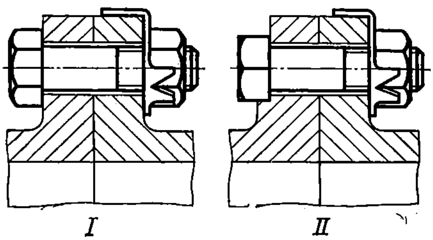


Fig. 581. Locking nuts with tab washers

Fig. 582, *I* and *II* illustrates incorrect settings of a tab washer. Here, the tab is bent against a semi-circular boss on the base; as the nut is untightened, the tab is shifted round the boss (as indicated by the arrow). A somewhat better setting is shown in Fig. 582, *III*, where the

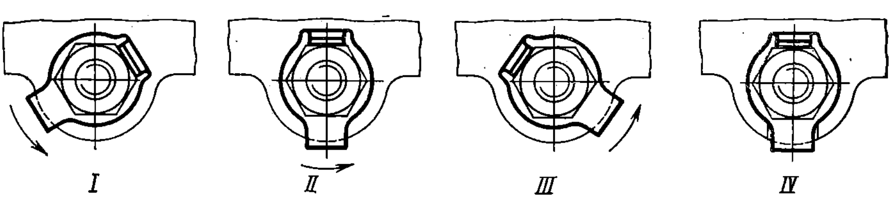


Fig. 582. Setting tab washer on part with half-round boss

washer cannot turn along with the nut because its tab runs up against the straight wall of the base. However, this arrangement is not free from drawbacks either, since the tab washer needs to be held against displacement during nut tightening. The proper setting is shown in Fig. 582, *IV*. Here, the tab is inserted into an inclin-

ed slot milled on the boss so that the washer is fixed against shifting in both directions.

Another example of tab washer setting is given in Fig. 583, *I*. The washer tab is bent against the peripheral surface of a cylindrical flange. Although the flange centre and the washer centre do not coincide, the washer does not provide positive locking since it is subject to turning round its axis through an angle sufficient to cause loosening of the nut. The proper setting can be ensured either by placing the tab into a suitable slot on the flange or (in order to save in machining) by making the tab washer bear against the cylindrical wall of the flange (Fig. 583, *II*).

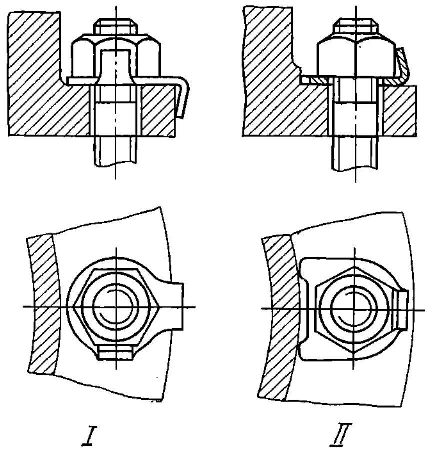


Fig. 583. Setting tab washers on cylindrical flange

I—incorrect; *II*—correct

A tab washer can be provided with a triangular recess in one of the tabs to allow the tab to be bent not only against a flat of the nut (Fig. 584, *I*) but also against a corner of its hexagon (Fig. 584, *II*). That makes it possible to lock the nut at every 30° of rotation rather than 60°.

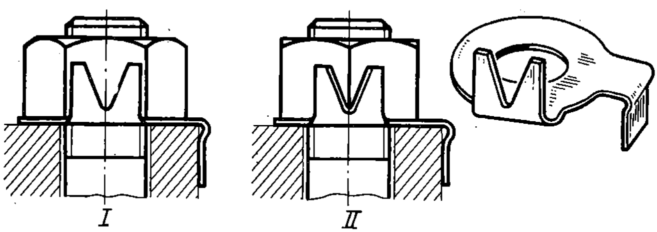


Fig. 584. Tab washers with recess

Fig. 585 shows a tab washer with a split tab; one half of the tab is bent back against the base and the other half, against a flat of the nut.

A variation of the tab washer with two internal tongues is shown in Fig. 586. The tongues fit into longitudinal slots on the bolt shank, and the tab is bent against one of the nut flats. The drawback to this design is that the bolt is weakened by the slots.

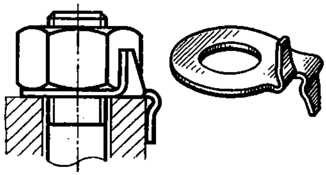


Fig. 585. Tab washer with split tab

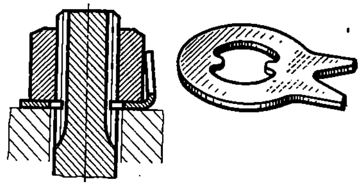


Fig. 586. Tab washer with internal tongues

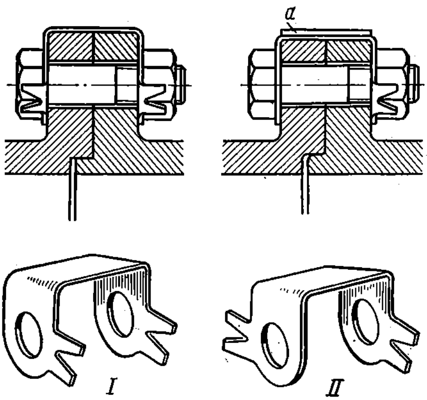


Fig. 587. Twin tab washers for nuts and bolts in flange assemblies

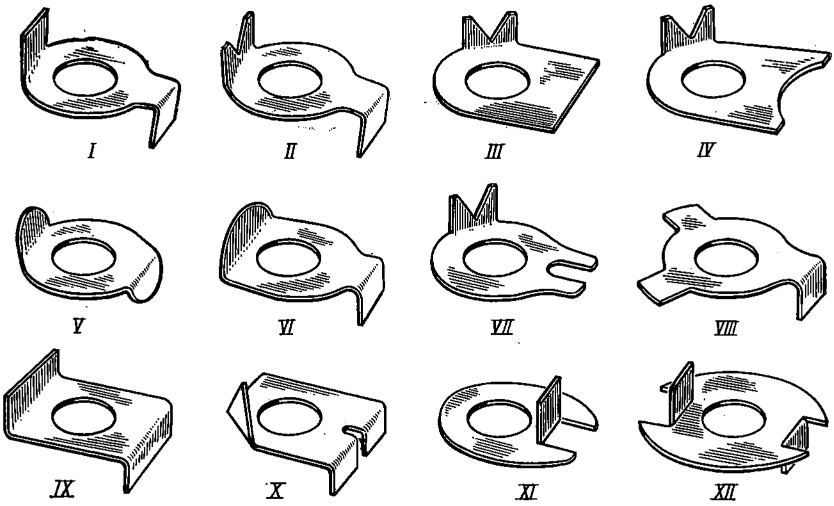


Fig. 588. Tab washers

A twin tab washer for nuts with bolts fastening cylindrical flanges is shown in Fig. 587. The washer (Fig. 587, *I*) is placed under the bolt head and the nut. The latter is locked by one of the tabs, and the washer itself is prevented from rotation as the other tab is bent against the bolt head. The method of locking presented in Fig. 587, *II* should be preferred because the washer is more positively fixed against rotation by being placed in slot *a* provided on the flanges.

Tab washers of various shapes (Fig. 588, *I-XII*) can be used, depending on application and on the method whereby they are fixed on the base. The washers are often made without tabs, shaped as round, oval, or rectangular strips (Fig. 588, *V*, *IX* and *X*). A washer of this type is fixed on the base and the nut by having its ends bent back; sometimes, tongues are formed on the ends, which makes possible stepless locking (Fig. 588, *XI* and *XII*).

Tab washers are disposable elements. The used washers are replaced with new ones after each disassembly.

8.4.2. Lock Washers

Locking with lock washers is based on constant friction forces created in the thread and on the bearing face of a nut. In this way the nut is fixed both to the bolt and to the base.

The friction forces prevent the nut from loosening under the action of vibration, pulsating forces, and also under residual strain developing within the joint (such as crushing of the bearing surfaces).

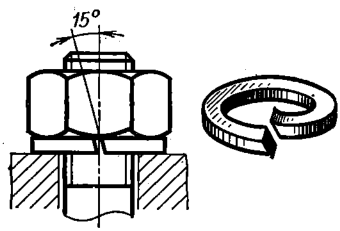


Fig. 589. Split spring washer

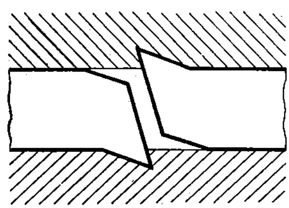


Fig. 590. Locking principle of split spring washers

Except for rare applications, lock washers are so installed that the nut can be fully tightened for positive connection with the base.

The simplest and most common type of lock washer (although far from being perfect) is a split spring washer (Fig. 589). It is essentially a ring made of hardened steel with a cut at an angle of about 15° to the ring axis. The cut has a 'left-hand' inclination for right-hand threads and a 'right-hand' inclination for left-hand threads. The ends of the washer at the cut are slightly separated and provided with sharp teeth. During tightening the spring washer is compress-

ed, and the teeth bite into the bearing surfaces of the nut and the base (Fig. 590), whereby the nut gets fixed to the base. The action of the teeth is markedly effective if the bearing surfaces are not too

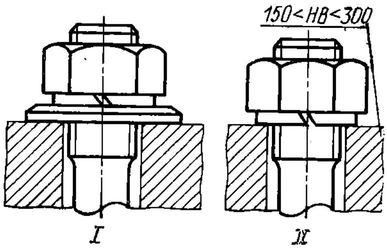


Fig. 591. Setting spring washers depending on hardness of base

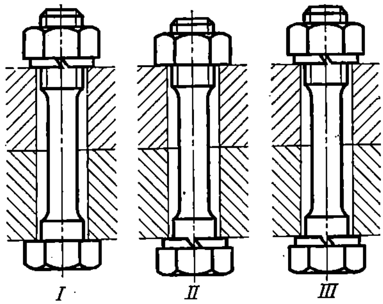


Fig. 592. Location of spring washers

hard (BHN $<$ 300). In the case of hard materials (hardened steel, nitrided steel, etc.), only friction forces are at work, and this reduces the locking effect.

Spring washers of this type are not for use on parts of metals with a hardness below BHN 150 (Fig. 591, *II*), e.g. castings of aluminium or magnesium alloys, since the teeth damage their surfaces. The use of plain steel washers underneath spring washers (Fig. 591, *I*) eliminates the action of the teeth on the base.

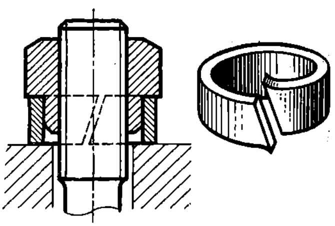


Fig. 593. Split spring washer of increased stiffness

The functioning of a bolt-and-nut assembly as an elastic system is not affected by the location of the spring washer; the latter may be placed under the bolt head (Fig. 592, *I*) as well as under the nut (Fig. 592, *II*). Two spring washers, one under the nut and the other under the bolt head (Fig. 592, *III*), double the elasticity of the assembly.

The principal disadvantage of a spring washer is that increase in the height or width of the washer cross section (Fig. 593) results in a greater force required to compress the washer, whereas its elastic deformation does not rise. Another serious disadvantage is that an initial tightening force is applied eccentrically, which is inevitable due to the fact that it is imparted to the nut and the base to a greater degree at the location of the teeth than at any other point along the circle.

In this respect, better performance is provided by Belleville spring washers, which are lock washers in the shape of a sphere (Fig. 594)

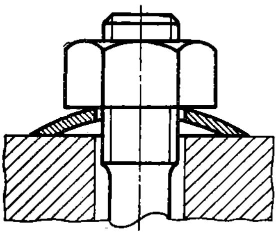


Fig. 594. Spherical lock washer

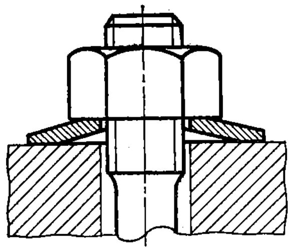


Fig. 595. Conical lock washer

or a cone (Fig. 595), by rectangular convex lock washers (Fig. 596), and by twisted lock washers (Fig. 597). A nut is tightened hard until

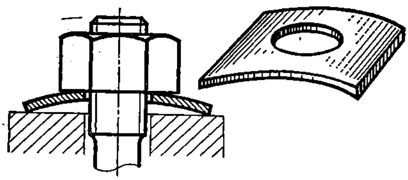


Fig. 596. Rectangular bowed lock washer

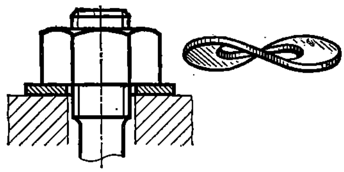


Fig. 597. Twisted lock washer

the lock washer underneath is flattened out and rigid contact with the bearing surface of the base is achieved. With the lock washer shown in Fig. 598, rigid contact is achieved when clearance s is eliminated.

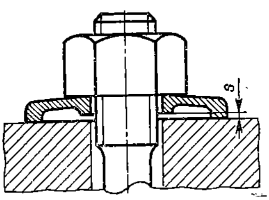


Fig. 598. Disc lock washer

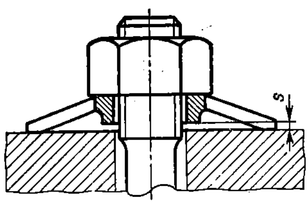


Fig. 599. Conical lock washer with radial slits

For greater elasticity, washers of this type can be provided with radial slits (Fig. 599) or with a cut (Fig. 600). In some applications, Belleville spring washers are placed both under the nut and the bolt head (Fig. 601).

For ease of assembly, lock washers are sometimes built in nuts (Figs. 602, 603) so that they can turn relative to each other.

Elasticity is further increased by special-form lock washers (Figs. 604, *I-IV* and 605). The washers of Fig. 604, *II*, *III*, *IV* and Fig.

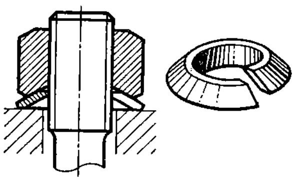


Fig. 600. Conical split spring washer

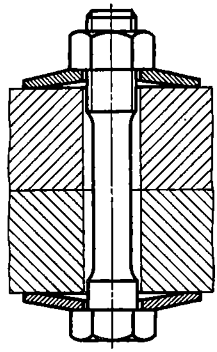


Fig. 601. Belleville spring washers placed under nut and bolt head

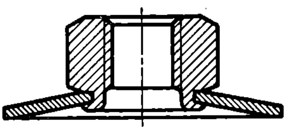


Fig. 602. Lock washer built into nut

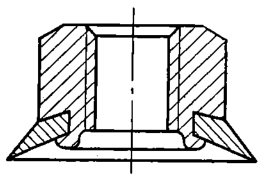


Fig. 603. Lock washer built into nut

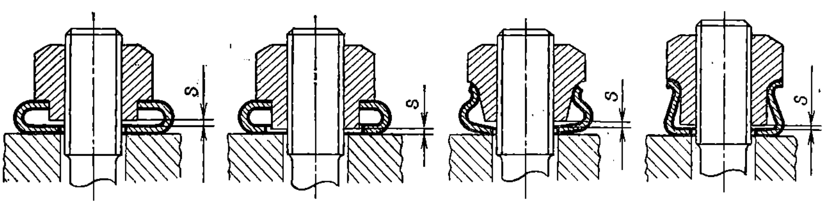


Fig. 604. Special-form lock washers

605 are used with tension nuts which feature a uniform load distribution among the threads.

In assemblies where still greater elasticity is required, use is made of stacked Belleville spring washers (Fig. 606), cylindrical (Fig.

607) or conical (Fig. 608) helical springs of rectangular section, or bellows-type springs (Fig. 609).

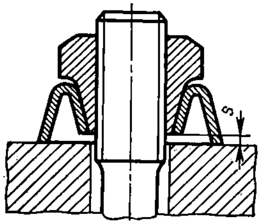


Fig. 605. Lock washer with conical seat for nut

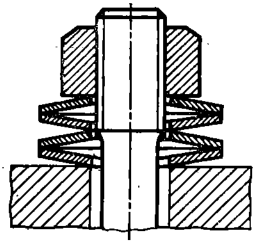
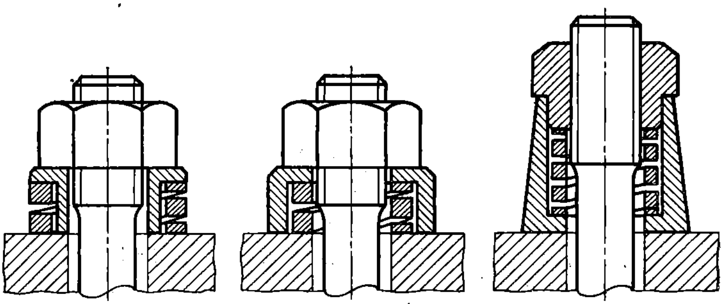


Fig. 606. Locking with stacked Belleville spring washers



I *II* *III*

Fig. 607. Locking with helical springs

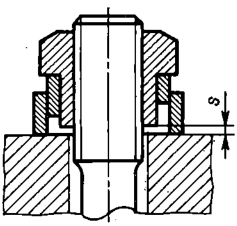


Fig. 608. Locking with conical spiral spring

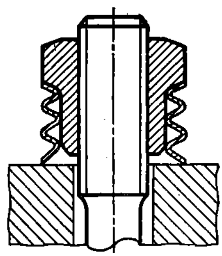


Fig. 609. Locking with bellows-type spring

In some applications, use is also made of conical spring rings (Fig. 610) in a single-ring (Fig. 610, *I*), double-ring (Fig. 610, *II*) or multiple-ring (Fig. 610, *III*) varieties.

The devices of Fig. 610, *I* and *III* are adapted for use with tension nuts. However, the skirt of such nuts is subjected to compression by the mating conical ring, which may neutralize the useful

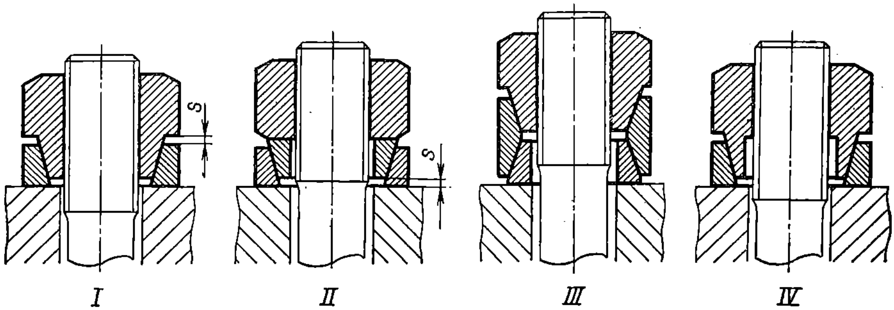


Fig. 610. Locking with conical spring rings

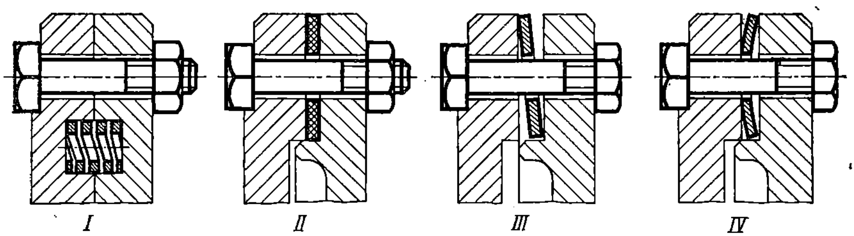


Fig. 611. Multiple locking of bolts on cylindrical flanges

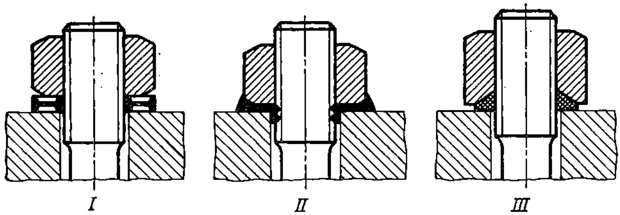


Fig. 612. Locking with plastic washers

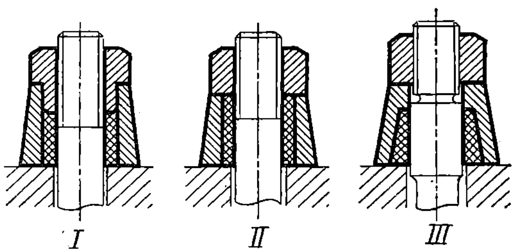


Fig. 613. Locking with plastic bushings placed in metal cups

effect of tension nuts. The remedy is to make a recess in the nut as shown in Fig. 610, *IV*.

In multiple-bolt assemblies (e.g. in fastening cylindrical flange-type components), elastic locking of the whole set of bolts can be

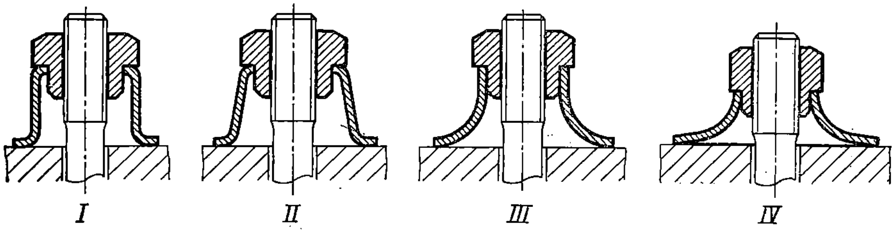


Fig. 614. Setting nuts on elastic spacers

effected by interposing between the flanges cylindrical helical springs (Fig. 611, *I*), elastic gaskets (Fig. 611, *II*) and circular elastic washers of various shapes (Fig. 611, *III* and *IV*).

Nuts can also be locked with elastic seal rings and bushings made from synthetic materials (nylon, capron, acrylon, etc.). Examples of such elements are given in Figs. 612 and 613.

A plastic ring of the shape shown in Fig. 612, *I* is placed under a nut. In the course of tightening, the ring is flattened out (Fig. 612, *II*) and its material is forced to flow into the threads. The nut becomes thereby locked and the whole joint sealed, which may be important for some applications. The seal ring shown in Fig. 612, *III* functions in much the same way.

Elastic bushings made of plastics (Fig. 613, *I*, *II* and *III*) are placed in metal cups. In their free state the bushings slightly jut out from the cups. In the course of nut tightening, the bushings are deformed, whereby the nuts are locked and the joints sealed.

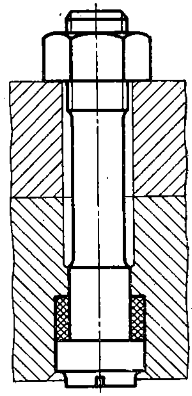


Fig. 615. Setting bolt on elastic bushing

In some instances, the nut is tightened up to bear against an elastic element which (unlike the arrangements just described) receives the full initial load and therefore should be sufficiently rigid and strong. Fig. 614, *I-IV* presents examples of such designs (in the order of increasing elasticity).

In the assembly illustrated in Fig. 615, an elastic bushing placed under the bolt head is fully loaded by the force of tightening. To prevent the squeezing of the bushing material from under the bolt head, the latter is inserted into the mating hole by a close sliding fit.

8.4.3. Tooth Lock Washers

These are washers made of hardened steel, provided with teeth, and placed under nuts (Figs. 616-618). The shape of the teeth is such that they do not impede the torquing of the nut but prevent its

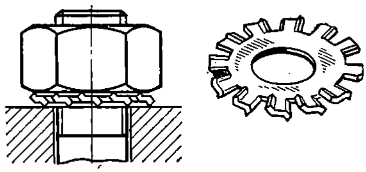


Fig. 616. Locking nut with external-teeth lock washers

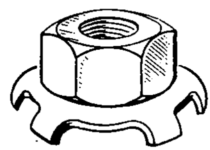


Fig. 617. Locking nut by convex lock washer with external teeth

backward rotation by biting into the nut and the base bearing faces and acting like retaining pawls in ratchet gearings.

In addition, the teeth or the washer as such are given a definite elasticity, so that the locking proves effective when the joint is sub-

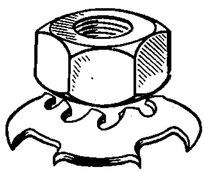


Fig. 618. Locking nut by convex lock washer with external and internal teeth

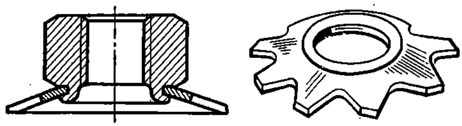


Fig. 619. Nut with tooth lock washer

ject to vibration and jarring or even when it slightly loosens. Therefore, tooth lock washers provide both elastic and positive locking; the nut is fixed to the bolt and the base.

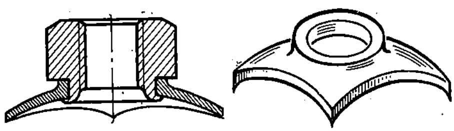


Fig. 620. Nut with square tooth lock washer

For ease of assembly, tooth lock washers are sometimes made complete with nuts (Figs. 619, and 620) so as to allow their relative rotation.

Tooth lock washers are not used for parts of soft metals (e.g. aluminium or magnesium alloys) or very hard metals (e.g. hardened

steel). With the former the washers damage the surfaces of the parts, whereas with the latter the effectiveness of locking is greatly reduced because the ratchet effect vanishes and only purely elastic forces

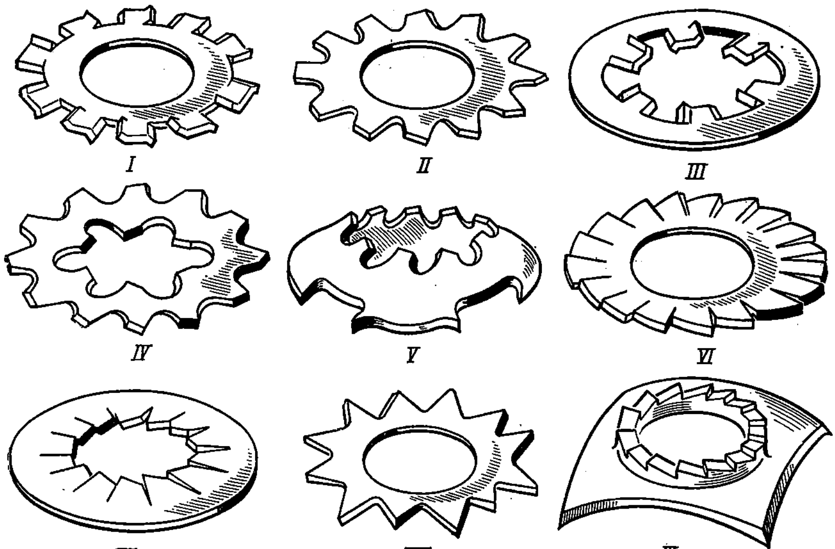


Fig. 621. Tooth lock washers

remain operative. Tooth lock washers are expedient to use for parts with hardness within a range of BHN 250 to 300.

Fig. 621, *I-IX* presents the most commonly used tooth lock washers.

8.5. Locking Plates

A locking plate is a strip of metal with a recess to receive the hexagon of a nut; it is placed about the nut and fastened to the base usually at two points (to prevent its displacement).

Various forms of locking plates are presented in Fig. 622, *I-IV*. The plate in Fig. 622, *IV* allows the nut to be fixed in a greater number of its angular positions.

The drawbacks of locking plates are the necessity to fasten these to the base and to lock the fasteners. For this reason, locking plates are used mainly for locking bolts placed in pairs where the locking plate can itself be locked by the nuts.

An elaborate method of locking designed for easy installation of a nut-locking device is shown in Fig. 623. The locking device comprises a washer with a hole to accommodate the nut hexagon and a

spring to hold the washer in place. The washer has tab a which serves to fix it to the base and tongues b which secure the spring.

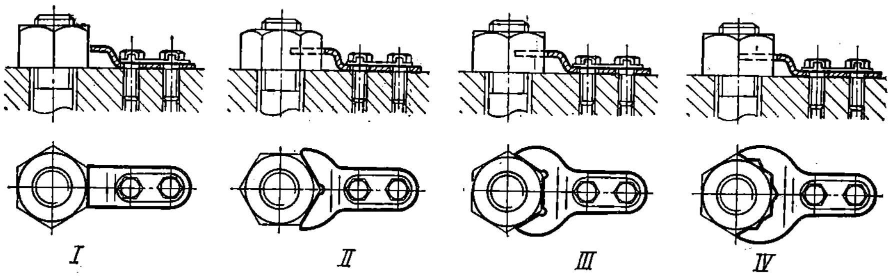


Fig. 622. Nuts fixed with locking plates

During assembly, the washer is put on the nut by compressing the spring which then springs back and holds the washer in position.

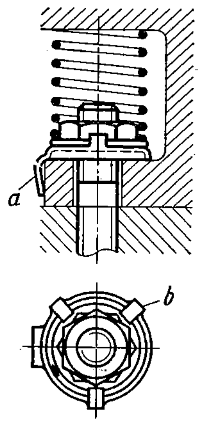


Fig. 623. Nut locked by safety washer pre-assembled with helical spring

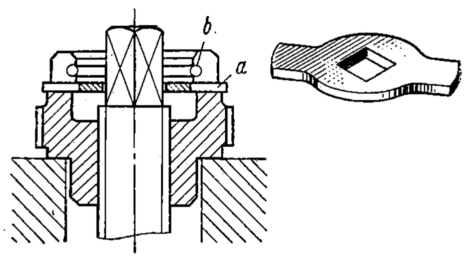


Fig. 624. Nut locked by contoured washer

A method for locking large bolts is illustrated in Fig. 624. The bolt shank is provided with a square on which lock washer a is placed so that its projections enter associated slots on the nut. The washer is fixed axially with snap ring b.

8.6. Wiring

In many applications nuts are locked by wiring. The wire is passed through one of the holes drilled on the flats of the nut being locked and through a hole in the adjacent nut or in any other nearby machine

element, or in an element specially introduced for the purpose (a screw, pin, etc.). The wire ends are bound together by stranding with pliers (Fig. 625).

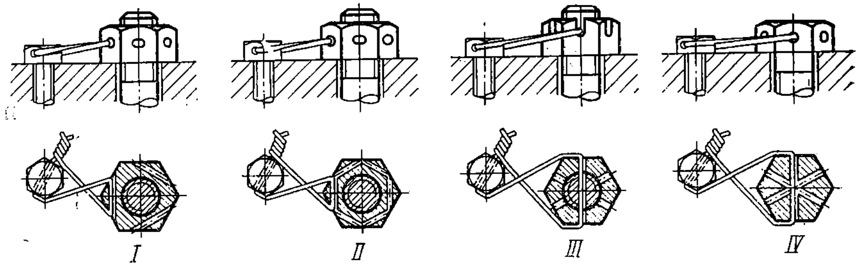


Fig. 625. Locking nuts by wiring

Wire holes in nuts are drilled with the aid of special jigs. Three holes are normally drilled (Fig. 625, *I*), and rarer, six holes (Fig. 625, *II*). A greater number of holes makes no sense, because proper wiring can be done with the nut angular position varying within a wide range. As distinct from most positive-locking methods, wiring makes it possible to fix the nut practically in any angular position.

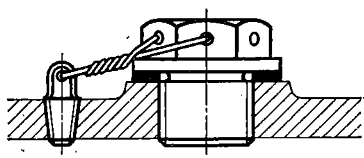


Fig. 626. Threaded plug locked by wiring

Fig. 625, *III* presents a method of wiring through the slots of a hexagon castle nut and the hole in the bolt. The method provides for locking the nut on the bolt and the bolt on the base.

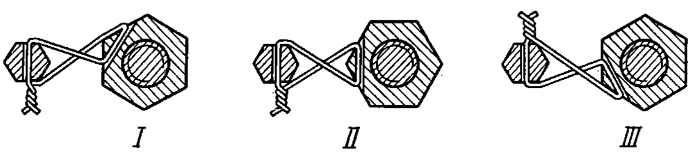


Fig. 627. Methods of wiring
I, *II*—incorrect; *III*—correct

Wire holes in bolt heads are usually drilled square to the flats (Fig. 625, *IV*). An instance of wiring a threaded plug is shown in Fig. 626.

The following rule must be observed in wiring: the tension developed in stranding the wire ends should create a moment which tends to tighten the nut being locked (Fig. 627, *III*). If the wire is stranded

in the opposite direction (see Fig. 627, *I*), it does not lock the nut; on the contrary, a moment develops which makes for its loosening.

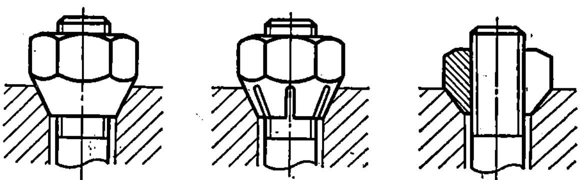
The locking also proves unreliable when the direction of wire tension is neutral, as shown in Fig. 627, *II*.

8.7. Self-Locking Nuts

Properly designed self-locking nuts should meet the following requirements:

(1) reliable locking;
(2) easy running up to the concluding stage of tightening;
(3) unimpeded full tightening;
(4) possibility of repeated use;
(5) adaptability to standard spanners;
(6) adaptability to mechanized assembly tools (powered nutrunners, etc.).

The action of self-locking nuts is generally based on the friction-locking principle, i.e. on the creation of increased friction in the



I *II* *III*

Fig. 628. Self-locking nuts with tapered bearing surfaces
I—solid; *II*—slotted; *III*—split

threads. Self-locking nuts which develop increased friction only at the end of tightening are the most advantageous. Designs of this kind utilize, to a greater or lesser degree, elastic properties of the nut material; therefore, nearly all self-locking nuts need to be heat treated.

The simplest way to increase friction is to use either interference-fit threads or threads of different pitch on the nut and the bolt. Torquing fasteners with interference-fit threads involves difficulties, and hence these are used in permanent assemblies (e.g. for setting studs in housings) or in assemblies where the nut must be locked in any position on the mating threaded shank.

Self-locking nuts with tapered bearing surfaces (Fig. 628), based on the principle of squeezing the bolt thread and developing increased friction on the bearing surfaces at the end of tightening are now seldom used for the following innate shortcomings:

(1) they require specially machined bearing surfaces in the base;
(2) they develop additional tensile stresses in the base and create the risk of crushing the bearing surfaces therein (especially with slow tapers);
(3) they make it impossible (especially with split tapers) to fully tighten the joint, because the thread in the region of the taper seizes on the bolt and gets jammed.

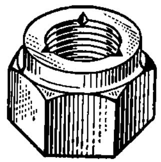


Fig. 629. Nut with deformed thread

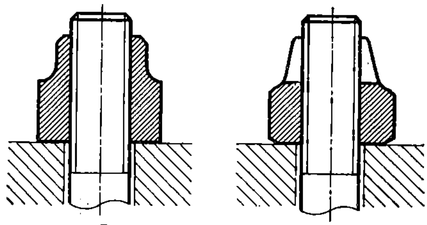


Fig. 630. Nuts with squeezed crowns
I—solid; *II*—slotted

A simple method for increasing friction in a thread at the end of tightening by deforming (e.g. notching with a centre punch) the last thread turns is shown in Fig. 629. However, this method is imperfect

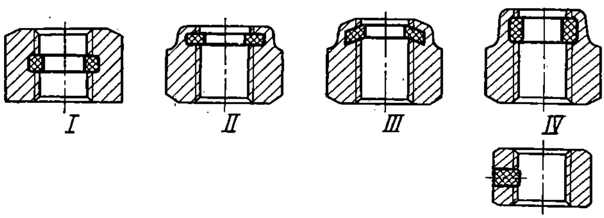


Fig. 631. Self-locking nuts with plastic inserts
I-IV—annular; *V*—pin-type

for lack of elasticity needed to maintain tightness in the thread in service conditions.

Fig. 630 shows self-locking nuts with elongated solid or split crowns which are squeezed when manufactured. As such a nut is run on the bolt, the crown produces increased friction in the thread. Nuts with split crowns (Fig. 630, *II*) provide a better locking effect due to greater elasticity of the crown segments.

Self-locking nuts with plastic inserts are shown in Fig. 631, *I—V*. When a nut of this type is screwed on a bolt, the latter forms threads

in the insert. Owing to its elasticity, the insert locks the nut in repeated mountings.

Fig. 632 presents self-locking nuts with elastic tapped collars. During manufacture, the collar is slightly upset after the nut has

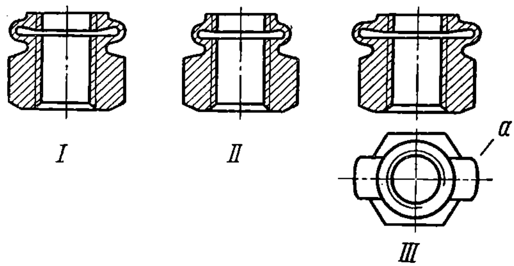


Fig. 632. Self-locking nuts with elastic collars
I—for open-ended spanner; *II*—for closed end spanner; *III*—nut with collar of increased elasticity

been tapped, so that the collar threads are axially displaced relative to the main threads of the nut. As the nut is screwed on a bolt, its end flexes the collar upwards, whereby spring pressure is developed in the thread.

The nut of Fig. 632, *I* can be handled with an open-ended spanner, whereas that of Fig. 632, *II*, with closed end spanner. Shown in Fig.

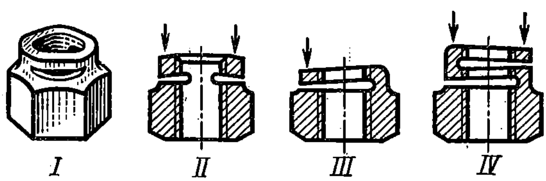


Fig. 633. Self-locking nuts
I—with three slits; *II*—with two slits; *III*—with a single slit; *IV*—with two staggered slits

632, *III* is a nut with a collar of increased elasticity; the collar is connected with the nut body by narrow arches *a*.

Fig. 633 presents self-locking nuts with slits made at the top, which is then radially squeezed or axially upset. The spring action of such a nut is similar to that just described.

The nuts shown in Fig. 634 are provided with crowns which are either upset or twisted relative to the nut main thread. Sometimes radial holes are drilled in the border area between the crown and the nut body for greater compliance.

Fig. 635 shows self-locking nuts whose principle is based on the well-known phenomenon wherein an elastic coil provides a grip-

ping action when turned on a shaft (this principle is used in some free-wheel designs).

A coil is formed at the top of the nuts by making slits as shown in Fig. 635. The coil remains connected with the nut body by a short

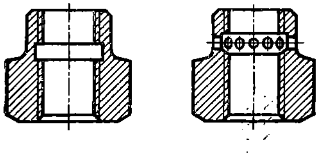


Fig. 634. Self-locking nuts with upset (*I*) and twisted (*II*) crowns

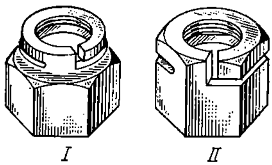


Fig. 635. Nuts with self-locking elastic coil
I—crowned nut; *II*—hexagon nut

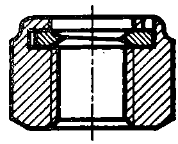


Fig. 636. Nut with self-locking elastic insert

portion, and its end is curved towards the nut centre to ensure some initial pressure. When the nut is screwed on, the coil does not impede its rotation; the backward rotation of the nut gives rise to the friction that locks it. A self-locking nut providing a similar action

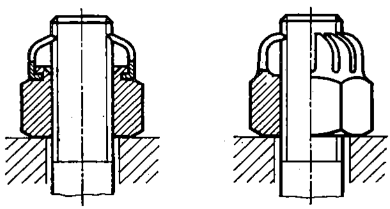


Fig. 637. Nuts with locking segments

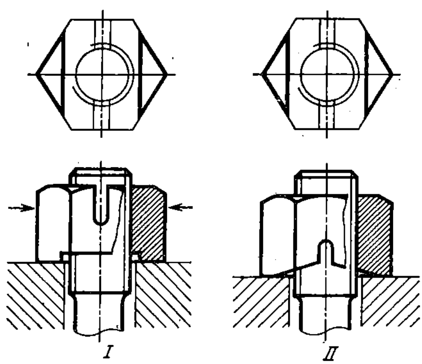


Fig. 638. Self-locking nuts squeezing on bolt from two sides in proportion to tightening force

is shown in Fig. 636. The difference is that here the spring coil is a separate element inserted into the nut body and held there by curling the edge of the bore.

Fig. 637 presents self-locking nuts with a spring-action feature comprising several segments whose ends are arranged in a helix and form a full turn of thread. The turn is either axially displaced with respect to the nut thread or drawn to the nut centre. In assembly, the bolt end flexes the segments outwards in the first case or

upwards in the second case, whereby spring pressure is developed in the screw joint.

Some varieties of self-locking nuts (Figs. 638, 639) are based upon a very expedient principle of developing in the thread an interference which is proportionate to the tightening force. That is achieved

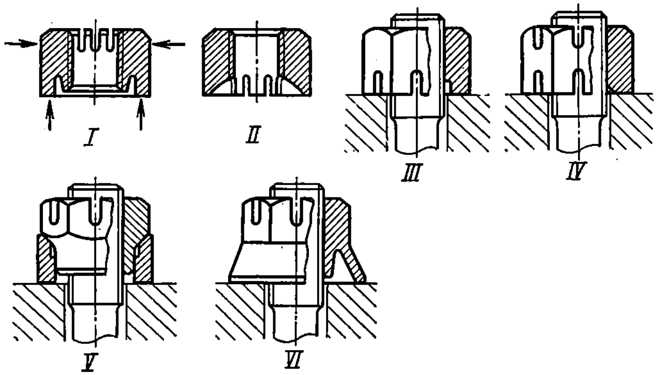


Fig. 639. Self-locking nuts

I—with circular recess and radial slots; *II*—with radial slots; *III*—with circular recess and radial slots on bearing face; *IV*—with radial slots on both end faces; *V*—with tapered bearing face; *VI*—with conical skirt and radial slots on top

by diverse methods. For instance (Fig. 638, *I*), a slot is machined on the bottom face of the nut so that it bears down against the base by two contact areas (defined by solid lines on the nut bottom view). Another slot is machined on the top face of the nut parallel to the bottom slot. In this way the nut virtually becomes divided into two halves which are connected by a bridge-like portion. The tightening force applied to the nut contact areas from below flexes the nut halves. The latter, acting as levers of the first kind, tighten up on the bolt, exerting a pressure which is proportionate to the tightening force. The nut of Fig. 638, *II* is similar in design.

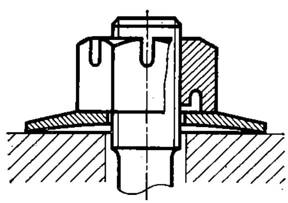


Fig. 640. Self-locking nut with a Belleville spring washer

With the nuts shown in Fig. 639, the threads in the upper region of the nut exert pressure on the bolt thread. The nut of Fig. 639, *I* has an annular bearing face and several slots in the top face. The load of tightening applied to the bearing face forces the slotted top segments against the bolt. The nuts in Fig. 639, *II*, *III* and *IV* operate on the same principle.

All the nuts in Figs. 638 and 639 have an additional merit, namely, they provide a uniform load distribution among the threads.

To maintain constant pressure against loosening under operating conditions involving vibrations and pulsating axial loads, it is necessary to use elastic bolts or additional elastic elements (Fig. 640).

Fig. 641 *I-IV* presents self-locking nuts whose action is provided by elastic elements made integral with the nuts. The nut of Fig. 641,

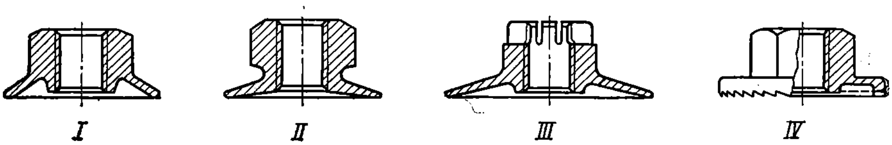


Fig. 641. Self-locking elastic nuts

III combines the principles of elastic locking with squeezing on the bolt thread upon tightening. The nut of Fig. 641, *IV* has teeth on the bearing face. The drawback to all the nuts of Fig. 641 is increased friction in tightening due to large-diameter bearing surfaces. Nuts with built-in elastic elements are much better in this respect.

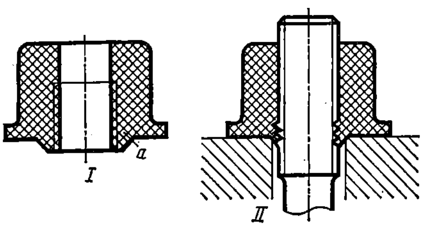


Fig. 642. Self-locking plastic nuts

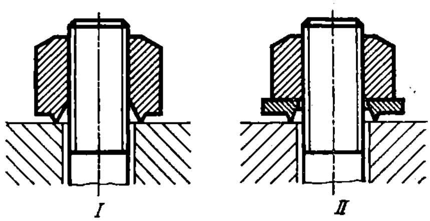


Fig. 643. Self-locking seal nuts for permanent joints

Fig. 642, *I*, *II* shows plastic (nylon) self-locking nuts designed for light loads. Such a nut is partly tapped as shown in Fig. 642, *I*. During the torquing of the nut, the bolt forms threads on the untapped portion of the hole, whereby the joint becomes tight. Additional tightness is achieved as the nut material at the tapered boss *a* flows into the bolt threads.

A self-locking nut for permanent joints with parts made of plastic materials is illustrated in Fig. 643, *I*. The bottom face of the nut is provided with a sharp ridge which forces the part material against the bolt threads and thereby ensures both the locking and sealing of the thread. The nut in Fig. 643, *II* is used with a washer which provides only thread sealing.

8.8. Locking Circular Nuts

Fig. 644 presents mostly outdated methods of locking circular nuts with thrust-type set screws. The set screw used as shown in Fig. 644, *I* tends to damage the thread; the introduction of inserts (Fig. 644, *II*, *III*) made of soft materials (e.g. bronze, plastics, etc.) eliminates the effect but complicates assembly and disassembly.

Another method which is also seldom used is shown in Fig. 645. Here, the nut is slotted, and the separated parts are drawn together

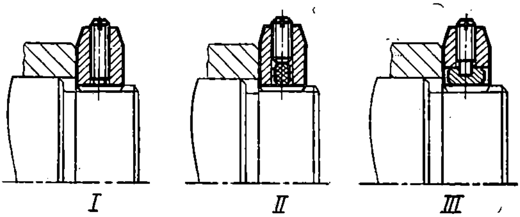


Fig. 644. Locking **circular nuts with set screws**
I—directly; *II*—through inserts of soft materials

(Fig. 645, *I*) or forced apart (Fig. 645, *II*) by means of a screw to create axial pressure in the thread. The nut is considerably weakened by the slot, which is the drawback to this method.

Spring nuts (Fig. 646) have gone out of use for the same reason. Sometimes they are still employed where the nut must be locked on the thread within a wide positional range.

The locking of circular nuts with a spring-action wire lock (Fig. 647) is also rarely used, since the lock cannot be fixed on the nut reliably enough. That disqualifies this method for use on high-speed shafts, where the lock may be displaced from its groove under the action of centrifugal forces.

The most commonly used methods of locking circular nuts with safety washers are presented in Fig. 648.

The safety washer is interposed between a part mounted on the shaft and the nut, which clamps the part. One of the washer tongues, which fixes the washer against rotation, engages a slot on the shaft, and another, a slot on the nut.

The safety washers are made from soft sheet steel 0.5 to 1.2 mm thick. They are disposable and replaced by new ones after each disassembly.

Two main methods for setting safety washers on shafts are employed. According to the first method, the tongue inserted in the shaft slot is bent away from the nut (Fig. 648, *I*), according to the second method it is bent under the nut (Fig. 648, *II*). The second method requires a slot of greater depth, and therefore the first method is preferred in practice.

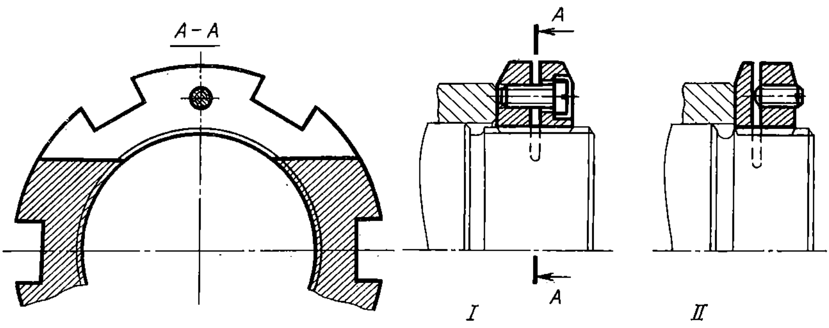


Fig. 645. Locking circular nuts by axial deformation

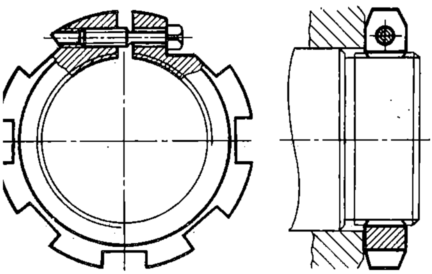


Fig. 646. Split spring nut

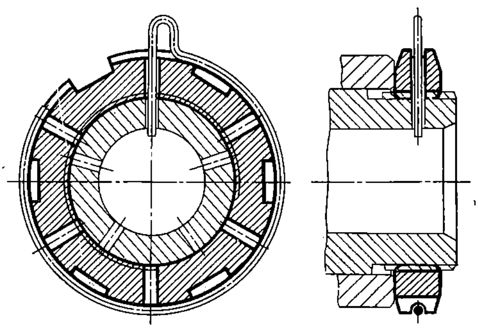


Fig. 647. Locking circular nut with wire lock

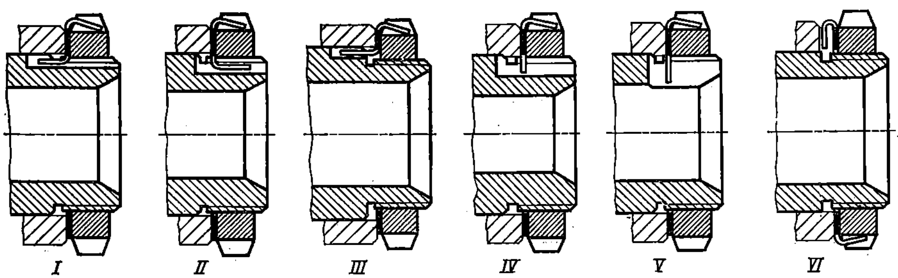


Fig. 648. Setting safety washers on shafts

I, III—tongue engages slot in hub; *II*—tongue engages slot in nut; *IV*—tongue engages deep slot in shaft; *V*—tongue engages a through slot in shaft; *VI*—tongue engages slot on hub end face

Where the shaft wall is thick enough to allow a deep slot, the associated tongue of the safety washer can be inserted therein unbent (Fig. 648, *IV*). That makes for more reliable locking.

Sometimes the slot is cut through as shown in Fig. 648, *V*, which substantially weakens the shaft. This is acceptable where the nut

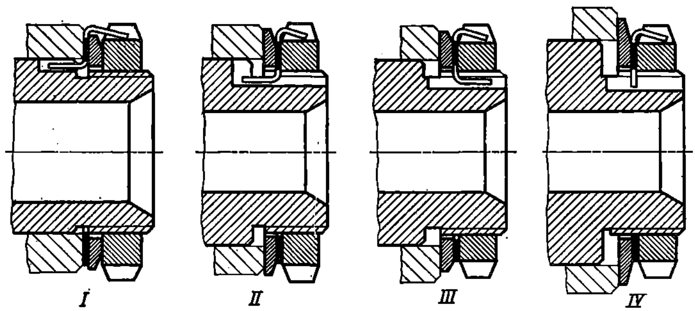


Fig. 649. Setting safety washers on stepped shafts using plain bearing washer

to be locked is situated on the shaft end, whereas the foregoing methods can be used for nuts placed in intermediate positions along the shaft, e.g. on stepped shafts.

Fig. 648, *VI* illustrates a method whereby the tongue of a safety washer is bent so as to engage a radial slot on a clamped part which

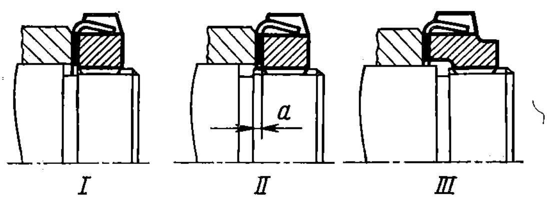


Fig. 650. Safety washer set on shafts
I—non-centralized; *II*—centralized on thread; *III*—centralized on journal

is itself fixed on a shaft by means of a key or splines. This method is fairly reliable and easy to effect.

Some methods for setting safety washers in combination with massive plain washers on stepped shafts are shown in Fig. 649, *I-IV*.

When installing a safety washer, it is incorrect to place it at the thread relief zone (Fig. 650, *I*) because the washer may get radially displaced. The washer should be located on the thread, which must extend by amount *a* beyond the nut (Fig. 650, *II*), or, better still, on the plain portion of the shaft (Fig. 650, *III*). In the latter case the nut must be recessed on the bearing face.

Where both a safety washer and plain washer are used, they must be centralized. Correct and incorrect settings are shown in Fig. 651.

The methods described above allow the nut to be fixed at angles equal to the angle between its slots. A finer angular fixing can be

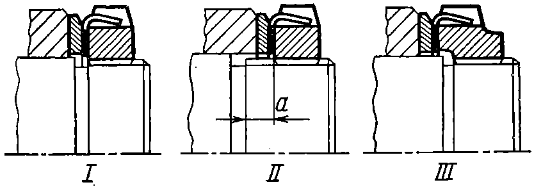


Fig. 651. Safety washers with bearing washers set on shafts

I—non-centralized; *II*—centralized on thread; *III*—centralized on journal

obtained by providing a safety washer with several tongues spaced at an angle which differs from the angle between the nut slots. The tongues can be arranged in a cluster (Fig. 652, *I*) or uniformly along

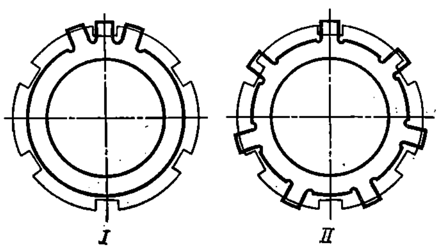


Fig. 652. Arranging tongues on safety washers

I—in cluster; *II*—along the periphery using the vernier effect

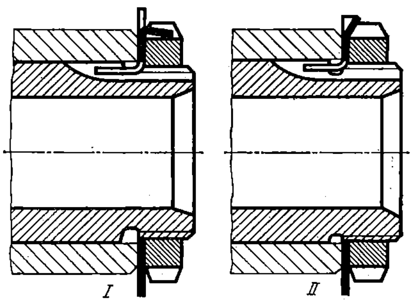


Fig. 653. Nut locked in any angular position

I—by tongue cut in round washer of increased diameter; *II*—by forcing washer edge into nut slot

the washer periphery (Fig. 652, *II*). In the latter case, the nut can be fixed in practically any angular position.

The nut can also be fixed in any angular position if the safety washer is made round and equal in diameter to the outward outline of unbent tongues (Fig. 653, *I*). The nut is tightened and the contour of its any suitable slot is layed out on the washer with a scriber. The assembly is then dismantled, and the washer is cut according to the layout to form the tongue. The nut with the washer is mounted on the shaft once again, and the tongue is bent into the nut slot. This method is slow, but it allows repeated use of the washer as well as locking in

any angular position of the nut. Another method used sometimes is to force portions of the washer into the nut slots (Fig. 653, *II*).

Some outlines of slots machined on shafts for safety-washer tongues are shown in Fig. 654, *I-IV*. The first slot should be preferred as the simplest to machine. The slot in Fig. 654, *III* takes more time to machine; in addition, the hole drilled for tool overrun greatly weakens the shaft. The flat shown in Fig. 654, *IV* cannot be recommended

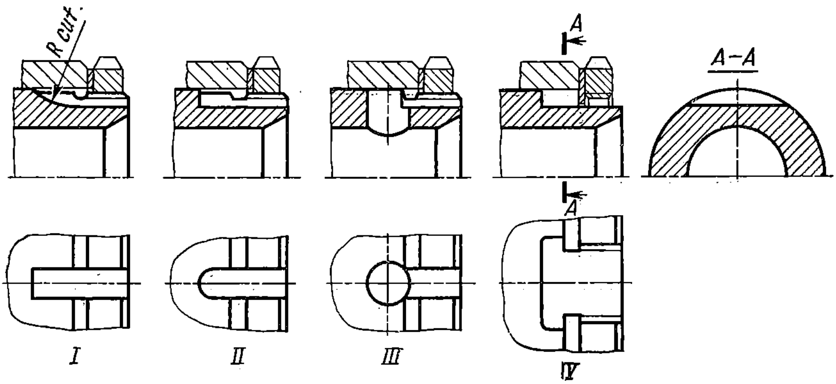


Fig. 654. Methods of machining shaft slots for safety washer tongues
I—milling by side milling cutter; *II*—milling by end milling cutter; *III*—shaping; *IV*—milling a flat instead of slot

since it weakens the shaft and fails to provide positive angular location of the tongue.

The drawback to locking with tongued washers is that the tongue engaging the shaft slot may be sheared off during nut tightening, with the failure remaining unnoticed. This danger is eliminated with the introduction of a rigid plain washer (Fig. 655) interposed between the nut and the safety washer and fixed against rotation. Such an arrangement is used in applications where reliability is crucial. In some applications the washer is provided with several tongues, and the shaft, with the same number of slots. Safety washers with internal teeth which engage shaft splines are safeguarded against failure.

Safety washers can be fixed on shafts by using some features of the assembled parts, e.g. keyways (Fig. 656) or grooves between splines (Fig. 657).

In the latter case, the splined portion of the shaft should be spaced from the lock nut so as to permit the accommodation of the safety washer tongue (distance *a* in Fig. 657, *I*), or one of the splines should be shortened by this length.

The best way to fix the safety washer is to provide it with internal teeth which engage the splines (Fig. 657, *II* and *III*).

Safety washers with external tongues which are bent back into the slots of nuts are not recommended for use on high-speed shafts, because the tongue may disengage from the slot under the action of

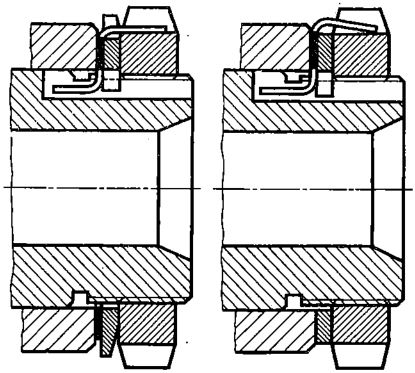


Fig. 655. Safety washer tongue is prevented from getting sheared off by introducing intermediate bearing washer

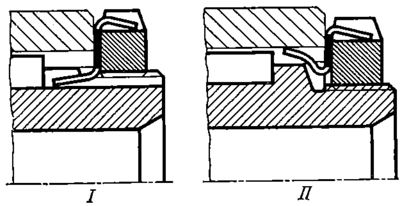


Fig. 656. Fixing safety washers
I—on splined shafts; *II*—in splined hubs

centrifugal forces. In such applications, circular nuts with radial slots on the outward face are preferred (Fig. 658, *II*), so that centrifugal force helps to keep the bent tongue in its place.

Reliable and easy-to-install cup-type safety washers (Fig. 659) have found application. Such washers are usually fixed on shafts by

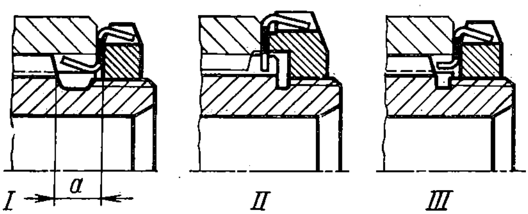


Fig. 657. Safety washers fixed on spline shafts
I—by tongue engaging in between adjacent hub splines; *II*—by internal teeth engaging shaft splines; *III*—by internal teeth engaging hub splines

means of internal teeth that engage directly the shaft splines or the splines of a hub mounted on the shaft. The cup of the washer encloses the nut; the washer material is forced into the nut slots by striking with a centre punch, whereby the nut is firmly secured. This method makes it possible to lock the nut in any angular position and to use the washer repeatedly. However, the nut here cannot be tightened

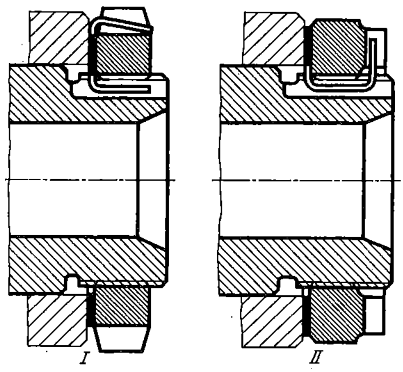


Fig. 658. Improper (*I*) and proper (*II*) setting of safety washers on high-speed shafts

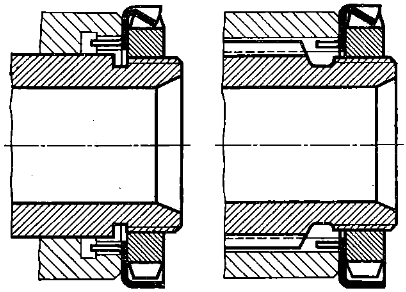


Fig. 659. Locking circular nuts with cup-type safety washers

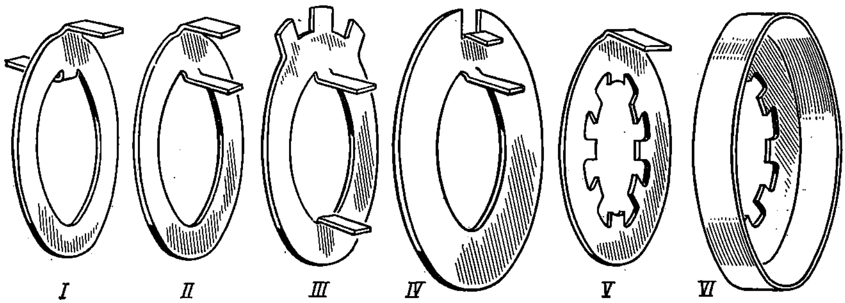


Fig. 660. Safety washers

I—with tongue bent away from nut; *II*—with tongue bent under nut; *III*—with several tongues; *IV*—with tongue layed out in situ and cut before final setting; *V*—with internal teeth for spline shafts; *VI*—cup-type with internal teeth for spline shafts

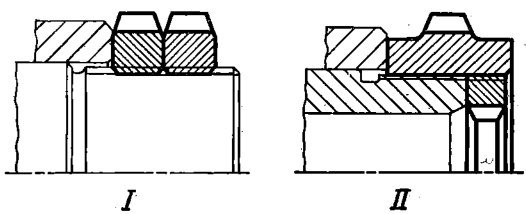


Fig. 661. Circular nuts fixed by

I—regular lock nut; *II*—male-threaded lock nut

with conventional hook-type or closed end spanners; only face spanners can be used for the purpose.

Fig. 660 presents different versions of tongued washers.

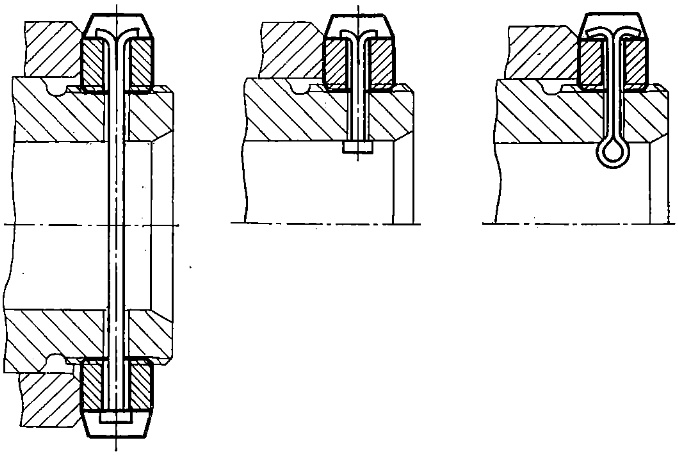


Fig. 662. Locking circular nuts with split cotter pins

Apart from tongued washers, some other devices can be used to lock circular nuts, e.g. lock nuts (Fig. 661), split cotter pins (Fig.

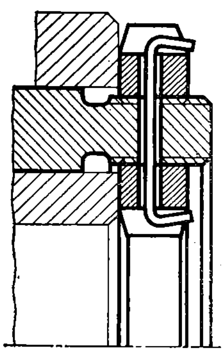


Fig. 663. Locking external and internal circular nuts with wire

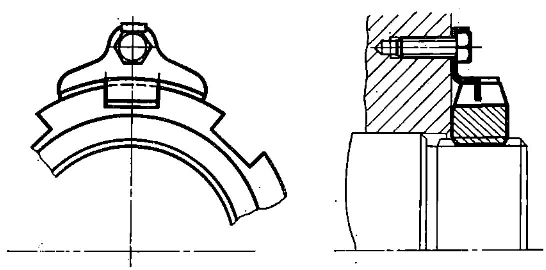


Fig. 664. Locking circular nut with locking plate

662), wires (Fig. 663), locking plates (Fig. 664), set screws (Fig. 665), pins (Fig. 666), spring-loaded locking devices (Fig. 667), wire binding (Fig. 668), etc.

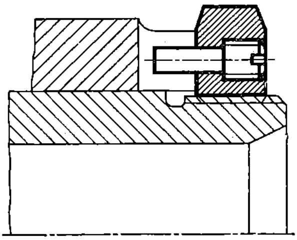


Fig. 665. Locking circular nut with set screw

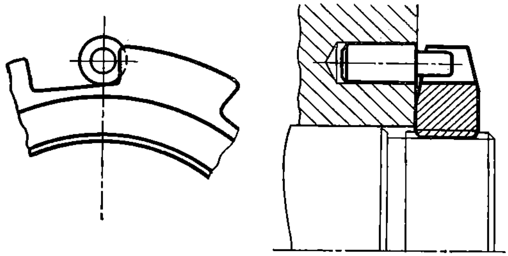


Fig. 666. Locking circular nut with pin (the nut is tightened up, the pin is set in place, and the nut is slightly turned back to overlap the pin shoulder)

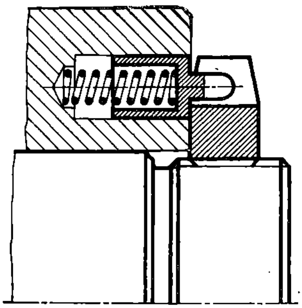


Fig. 667. Locking circular nut with spring plunger (the plunger is sunk by face spanner during nut torquing)

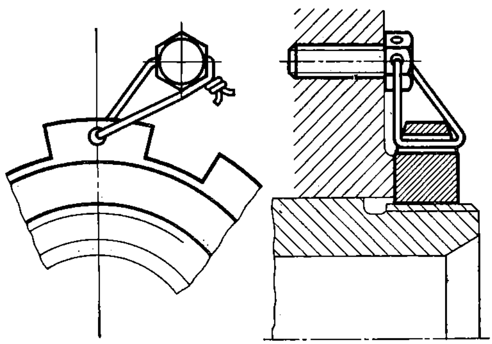


Fig. 668. Circular nut locked by wire binding

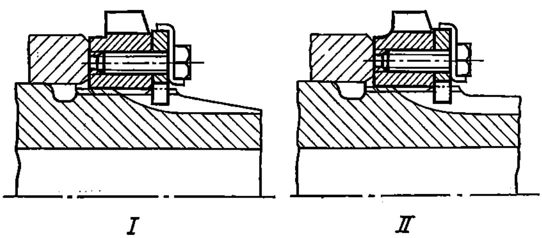


Fig. 669. Locking circular nuts by screwed-on washers having a tooth which engages in slot or groove between splines on shaft

Fig. 669 illustrates methods which are often used for locking circular nuts in intermediate positions on stepped shafts. Here, use is made of washers with a tooth inserted into a slot (Fig. 669, *I*) or groove between splines (Fig. 669, *II*) on the shaft. The screws which fasten the washers to the nuts are themselves locked with tab washers.

A method for friction locking of a circular nut by means of a spring is illustrated in Fig. 670. In the assembly shown in Fig. 671, a circular nut is positively locked by spring-loaded member *a* whose

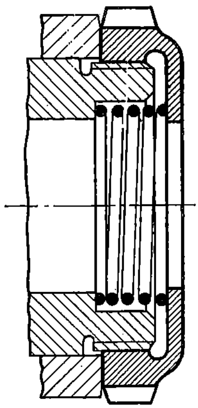


Fig. 670. Friction-locking of circular nut by spring

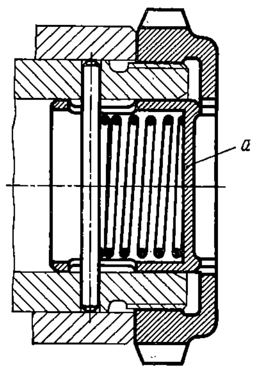


Fig. 671. Circular nut locked by toothed spring plunger

teeth engage the mating teeth in the nut body. The nut is handled with a special spanner which pushes the locking member back for disengagement to make it possible to turn the nut.

Locking a circular nut with a conical threaded plug is shown in Fig. 672. When screwed in, the plug expands the split end of the shaft to lock the nut.

Fig. 673 illustrates a special method for locking, where the nut is provided with a thin flange. To lock the nut, the flange material is forced into a slot on the adjacent hub by striking with a centre punch.

An ingenious method of locking circular nuts of increased length is presented in Fig. 674, *IV*. A special-form lock washer *a* made of soft steel is set on the shaft end face. Tongues *b* of the washer engage slots on the shaft end face and peripheral teeth *c* engage grooves in the circular nut. The washer is fixed by lock nut *d* with inclined slots *e*, which itself is locked in position by bending the washer's outward edges into the inclined slots.

Circular nuts with end-face slots (Figs. 675-678) are locked as described above.

Fig. 679 shows the locking of a circular nut mounted in an intermediate position on a stepped shaft. The nut is fixed by sleeve *a*

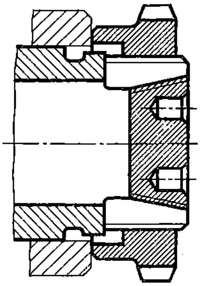


Fig. 672. Circular nut locked by conical threaded plug

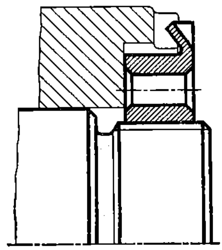


Fig. 673. Flanged nut locked by forcing its flange into adjoining slot

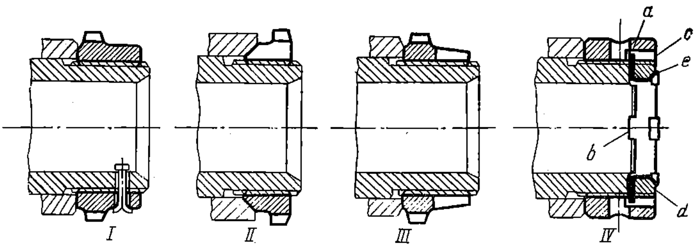


Fig. 674. Methods for locking circular nuts of increased length
I—by cotter pin; *II*, *III*—self-locking by spring action; *IV*—by lock washer *a* and lock nut *d*

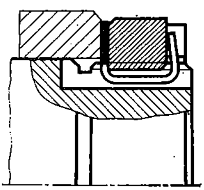


Fig. 675. Circular nut with end-face slots locked by tongued washer

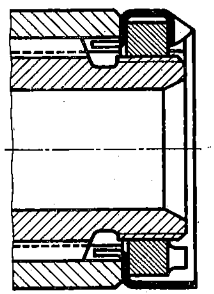


Fig. 676. Circular nut with end-face slots locked by cup-type safety washer

provided with teeth which engage slots on the nut face and slots (Fig. 679, *I*) or splines (Fig. 679, *II*) on the shaft. The sleeve is axially fixed on the shaft by adjoining hub *b*.

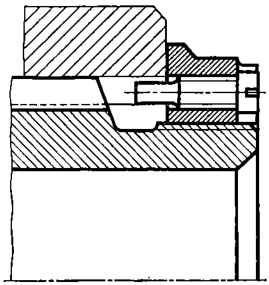


Fig. 677. Circular nut with end-face slots locked by screw

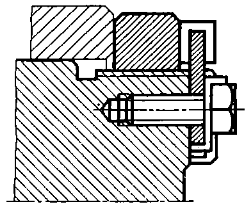


Fig. 678. Circular nut with end-face tooth fixed by locking plate

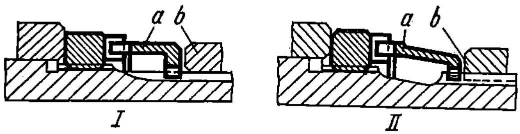


Fig. 679. Circular nut with end-face slot locked on stepped shaft by toothed sleeve engaging nut slots (*I*) and shaft splines (*II*)

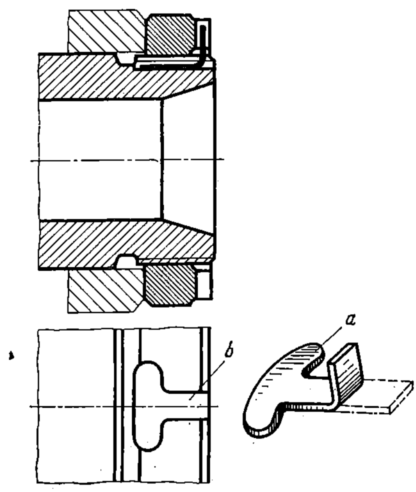


Fig. 680. Circular nut with end-face slots locked by T-shape sheet-metal strip

An unusual locking method is presented in Fig. 680. The locking element *a* is a T-shape strip of soft sheet metal. It is placed into a T-shaped slot *b* of small depth made on the shaft; its free end is bent back to engage a slot on the nut end face. This arrangement has

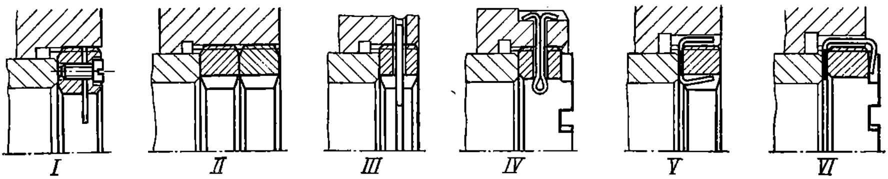


Fig. 681. Locking male-threaded circular nuts

I—by screw; *II*—by lock nut; *III*—by elastic retaining ring; *IV*—by cotter pin; *V*—by safety washer; *VI*—by safety washer (tongue bent back into nut end-face slot)

a serious drawback; unlike tongued washers, the locking element here is not tightened by the nut. Lest it should shift in the slot, the locking element must be close-fitted in longitudinal part *b* of the slot.

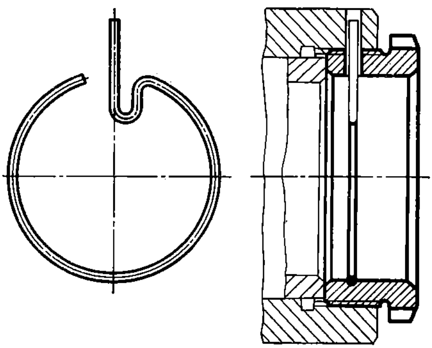


Fig. 682. Nut locked by retaining ring

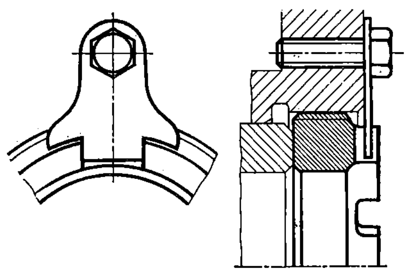


Fig. 683. Nut with end-face slots secured with locking plate

Male-threaded circular nuts (Figs. 681-686) are locked similarly to plain circular nuts.

For this kind of nuts, locking with a retaining ring (Fig. 682) can be used on shafts rotating at extremely high speeds. Of all the locking methods shown in Fig. 681, *I-VI*, the most commonly used is locking with tongued washers (Fig. 681, *V* and *VI*).

Nuts with end-face slots are often fixed with locking plates (Fig. 683). The locking plates themselves must be fixed against rotation relative to the fastening screw, which is effected by bringing them to bear against the periphery of the slotted portion of the nut, as in Fig. 683, or by some other method. In some applications, self-locking spring-action nuts can be found (see Fig. 684).

Fig. 685 shows the locking of cap nuts (often used in mechanical engineering) by means of cup-type safety washers whose edges are forced into the nut slots. The washers are fixed against rotation with the aid of teeth which engage the splines of the hub being locked.

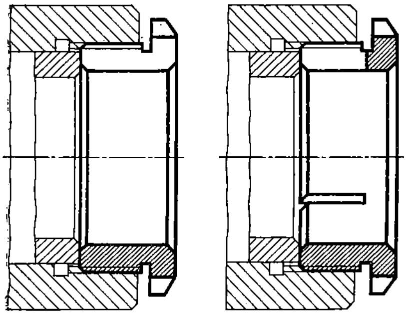


Fig. 684. Self-locking elastic sleeve nuts

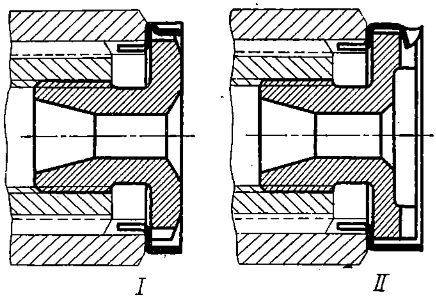


Fig. 685. Nuts locked by cup-type safety washers

I—nut with peripheral slots; *II*—nut with end-face slots

Nuts of this type can be secured by spring-loaded locking devices which engage the elements whereby the nuts are torqued (hexagon in Fig. 686, *I* or serrations in Fig. 686, *II*). To tighten or remove

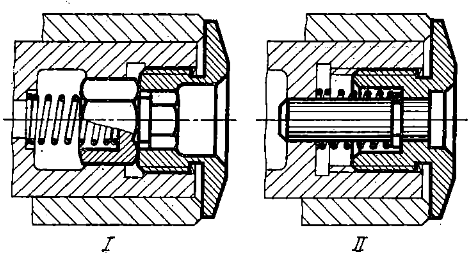


Fig. 686. Nuts fixed by spring-loaded locking devices

I—nuts with hexagon socket; *II*—nuts with serrated socket

the nut, a key spanner inserted into the nut socket pushes the lock pin and so releases the nut for turning. After tightening, the spring-loaded lock pin snaps back into the nut and locks it.

8.9. Locking Cap Screws

Hexagon cap screws are locked by the same methods as used for locking nuts, i.e. by means of spring washers (Fig. 687, *I*, *II*), tooth lock washers (Fig. 687, *III*), tab washers (Fig. 687, *IV*),

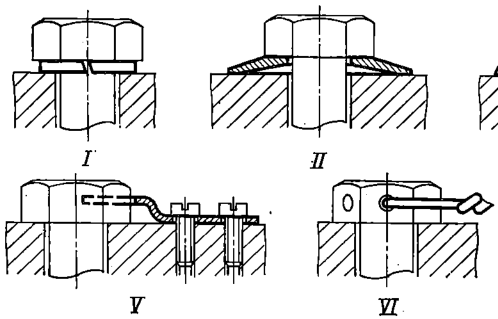


Fig. 687. Locking cap screws
I—by split spring washer; *II*—by Belleville spring washer; *III*—by tooth lock washer; *IV*—by tab washer; *V*—by locking plate; *VI*—by wire binding

locking plates (Fig. 687, *V*), wiring (Fig. 687, *VI*), etc. Fig. 688, *I*, *II* gives instances of locking a cap screw turned into a shaft end face by the aid of spring-action lock washers.

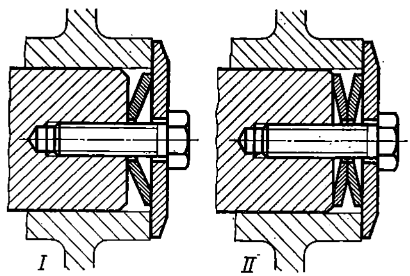


Fig. 688. Cap screws turned into shaft end and locked by Belleville spring washers

Cap screws can also be locked at the shank with the use of split cotter pins (Fig. 689, *I*), elastic inserts (Fig. 689, *II*), etc. A method whereby increased friction in the thread is developed by upsetting the end portion relative to the main part of the threaded shank is illustrated in Fig. 689, *III*. Where the end of the screw can be allowed to extend from the fastened part, the screw is locked by a lock nut (Fig. 689, *IV*), a lock nut in combination with a spring washer (Fig. 689, *V*), a castle nut with a split cotter pin (Fig. 689, *VI*), etc.

Locking large cap screws is shown in Fig. 690. Here, the screw is locked by turning a tapered screw plug into its split end (Fig. 690, *I*) or by driving therein a plain taper plug fixed by a retaining ring (Fig. 690, *II*).

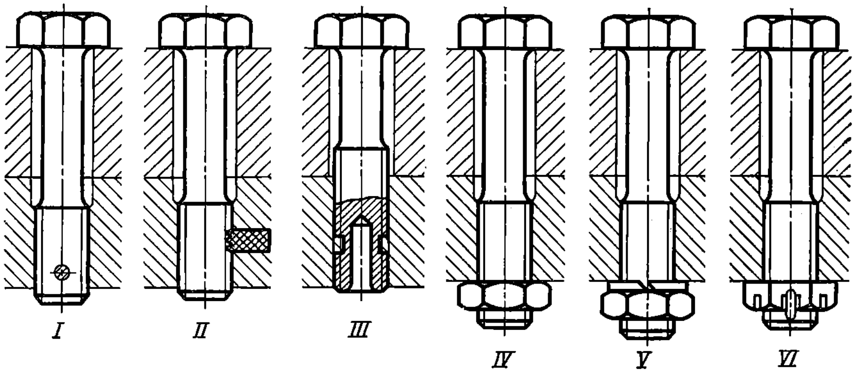


Fig. 689. Locking cap screws at threaded shank
I—by cotter pin; *II*—by elastic insert; *III*—by upsetting shank end; *IV*—by lock nut; *V*—by lock nut with spring ring; *VI*—by slotted nut with cotter pin

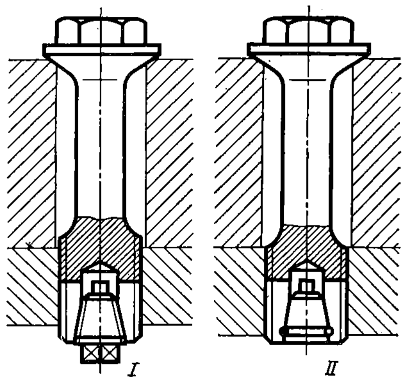


Fig. 690. Locking cap screws with split end
I—by threaded conical plug; *II*—by plain conical plug

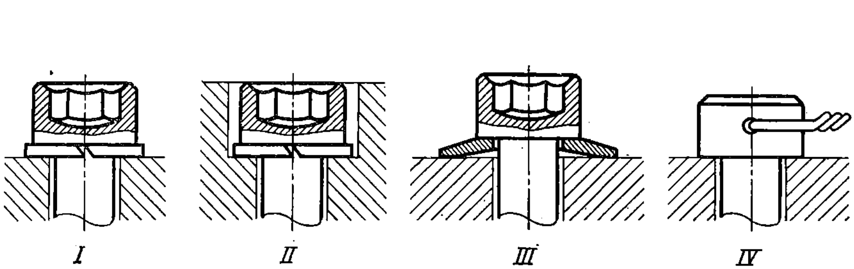


Fig. 691. Locking hexagon-socket cap screws

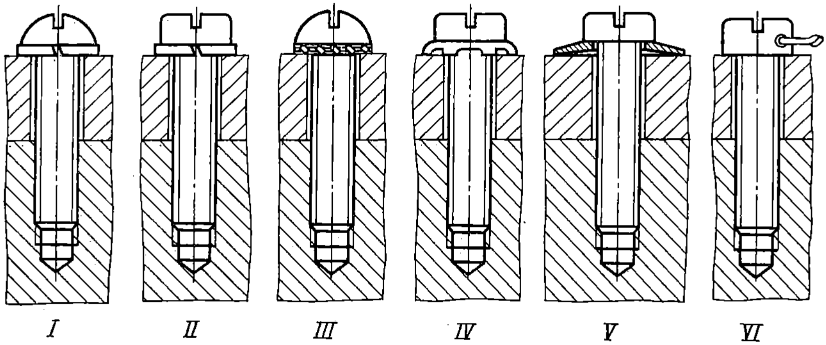


Fig. 692. Locking round and cheese head machine screws

I, *II*—by spring washers; *III*, *IV*—by tooth lock washers; *V*—by Belleville spring washer; *VI*—by wire binding

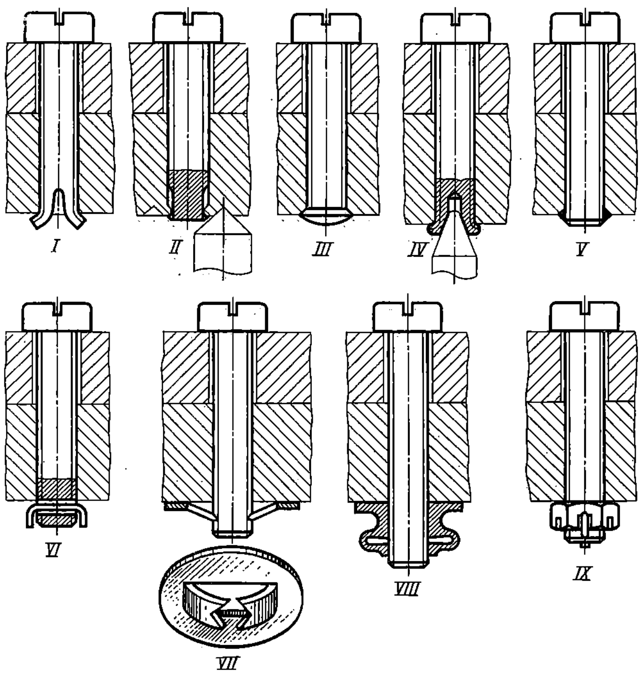


Fig. 693. Locking machine screws at threaded shank

I—by clinching split end; *II*—by notching metal round shank end; *III*—by peening shank end; *IV*—by staking shank end; *V*—by welding (brazing) shank end; *VI*—by cotter pin; *VII*—by pronged spring nut; *VIII*—by spring nut; *IX*—by castle nut with cotter pin

Hexagon socket head cap screws are much more difficult to lock. Such screws can be fixed by the head using only elastic-locking methods (Fig. 691, *I*, *II*, *III*) and wiring (Fig. 691, *IV*), and by the shank using all the methods shown in Fig. 689.

8.10. Locking Machine Screws

Machine screws with slotted round heads and cheese heads are readily locked with spring washers and tooth lock washers (Fig. 692,

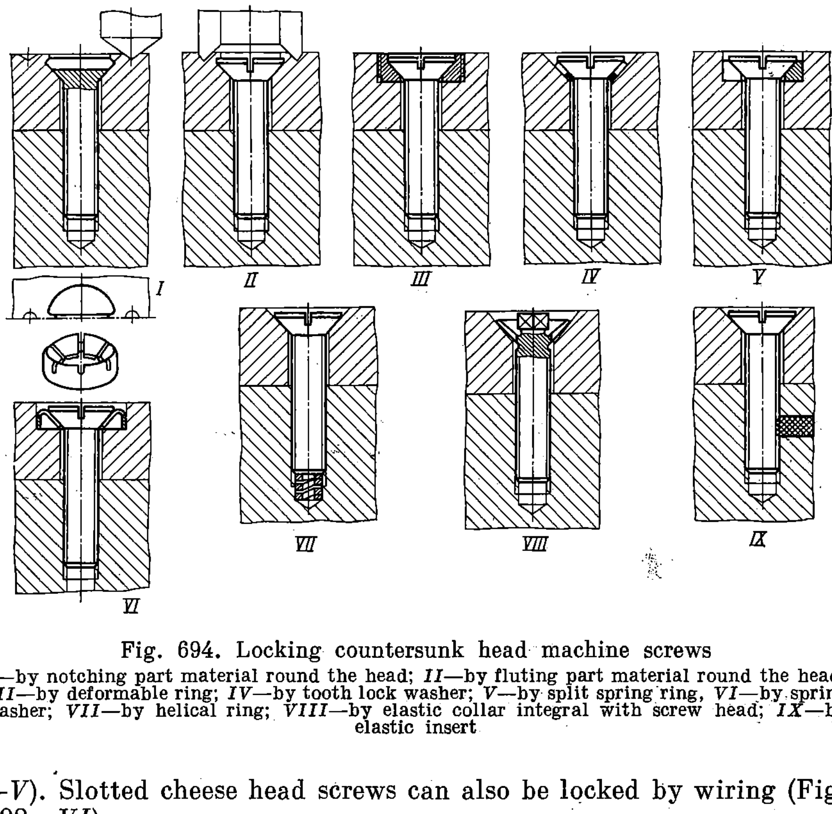


Fig. 694. Locking countersunk head machine screws

I—by notching part material round the head; *II*—by fluting part material round the head; *III*—by deformable ring; *IV*—by tooth lock washer; *V*—by split spring ring, *VI*—by spring washer; *VII*—by helical ring; *VIII*—by elastic collar integral with screw head; *IX*—by elastic insert

I-V). Slotted cheese head screws can also be locked by wiring (Fig. 692, *VI*).

Machine screws set in through holes can be locked at the shank end (Fig. 693).

Countersunk head screws are rather difficult to lock; a universal method of locking such screws has yet to be developed.

Fig. 694 illustrates methods for locking countersunk head machine screws set in blind tapped holes. The methods in Fig. 694, *I-III* provide for permanent locking. According to the method in

Fig. 694, *III*, a soft-metal ring is placed under the screw head; as the screw is tightened, the ring is squeezed, its metal flows upwards and seals the screw head. The ring is fixed against rotation by serrations on the counterbore wall.

An instance of locking by a tapered tooth lock washer (see Fig. 695) is given in Fig. 694, *IV*. In the design of Fig. 694, *V* a split

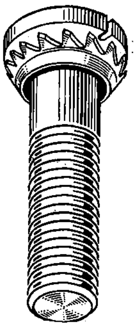


Fig. 695. Screw with tooth lock washer

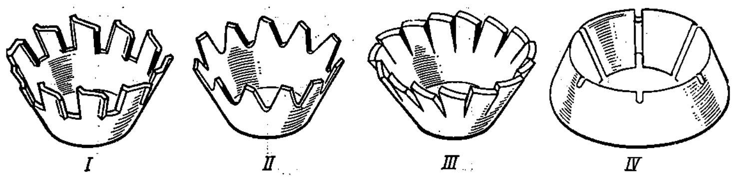


Fig. 696. Spring and tooth lock washers for countersunk head machine screws

spring ring with a conical bearing surface expands when the screw is tightened, so that the ring gets jammed between the screw head and the counterbore. This is the simplest and most effective method of screw locking.

Tooth lock washers and spring washers which have found application for locking countersunk head screws are shown in Fig. 696, *I-IV*.

8.11. Locking Miscellaneous Parts

Locking Hollow Screws. Methods for locking hollow screws, fairly often used in engineering, are presented in Fig. 697, *I-IV*. The method in Fig. 697, *VI* makes use of a spring-loaded lock pin *a* which enters a serrated socket in the screw and simultaneously engages a square hole in the base. To release the screw for turning, the lock pin is pushed downwards.

Hollow screws which hold down various inserts are locked by the methods illustrated in Fig. 698.

Where a hollow screw is to be fixed in any required axial position, it is locked by the methods shown in Fig. 699, *I*, *II*. Here, locking is effected by expanding the slotted portion of the screw by means of a taper element.

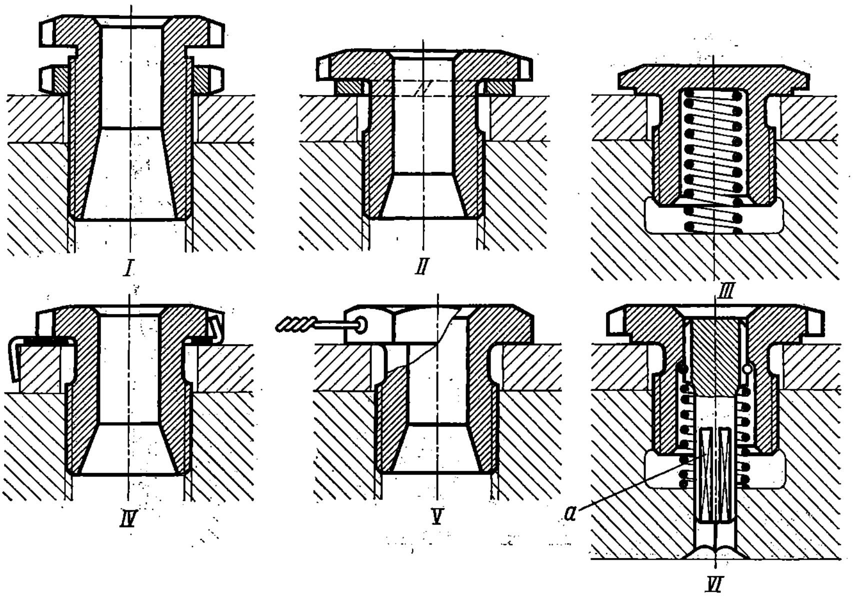


Fig. 697. Locking hollow screws

I—by lock nut; *II*—by spring washer; *III*—by helical spring; *IV*—by tab washer; *V*—by wire binding; *VI*—by spring-loaded lock pin

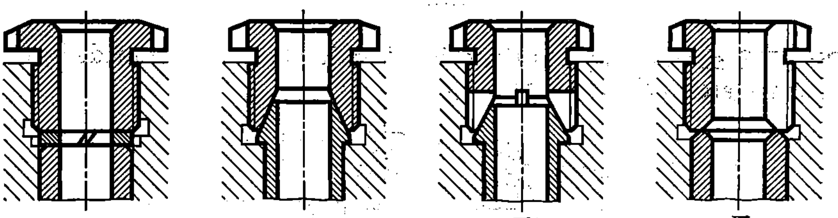


Fig. 698. Locking hollow screws which hold down internal parts

I—by spring ring; *II*—by taper socket on screw end; *III*—by split taper socket on screw end; *V*—by split elastic screw

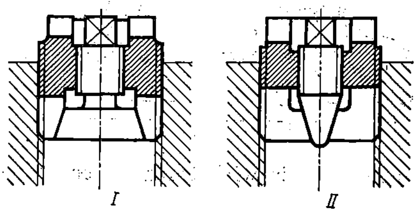


Fig. 699. Split-end hollow screws locked by taper inserts in any axial position

Locking Pipe Connections. The most common method of locking pipe connections is wiring. Examples are given in Figs. 700-702.

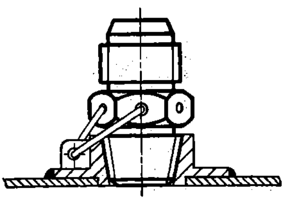


Fig. 700. Conical thread fitting locked by wiring

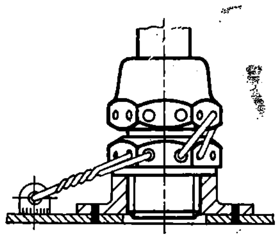


Fig. 701. Union-type fitting locked by wiring (fitting is bound to base and union nut to the fitting)

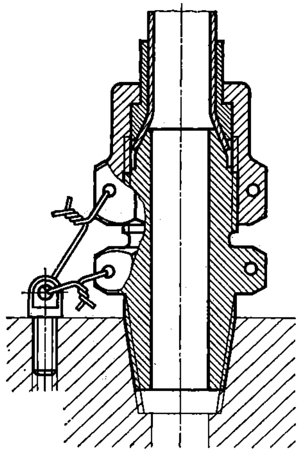


Fig. 702. Union-type fitting locked by wiring (both fitting and union nut are bound to base)

Fig. 703 shows a hydraulic control unit where all threaded components are locked by wiring.

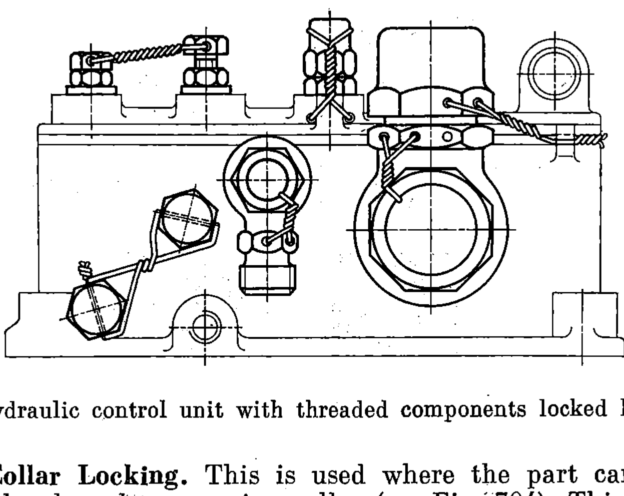


Fig. 703. Hydraulic control unit with threaded components locked by wiring

Spring-Collar Locking. This is used where the part carrying a screw can be shaped as a spring collar (see Fig. 704). This method

provides locking of the screw in any position and therefore is often used for threaded push rods in linkages, lever-type drives, etc.

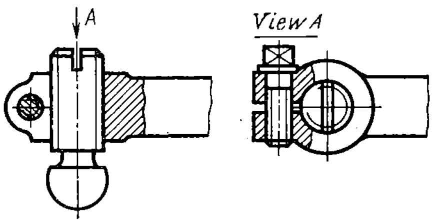


Fig. 704. Screws locked by spring collars

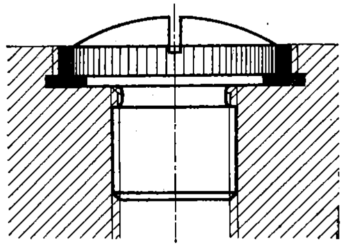


Fig. 705. Locking by deformable plastic insert

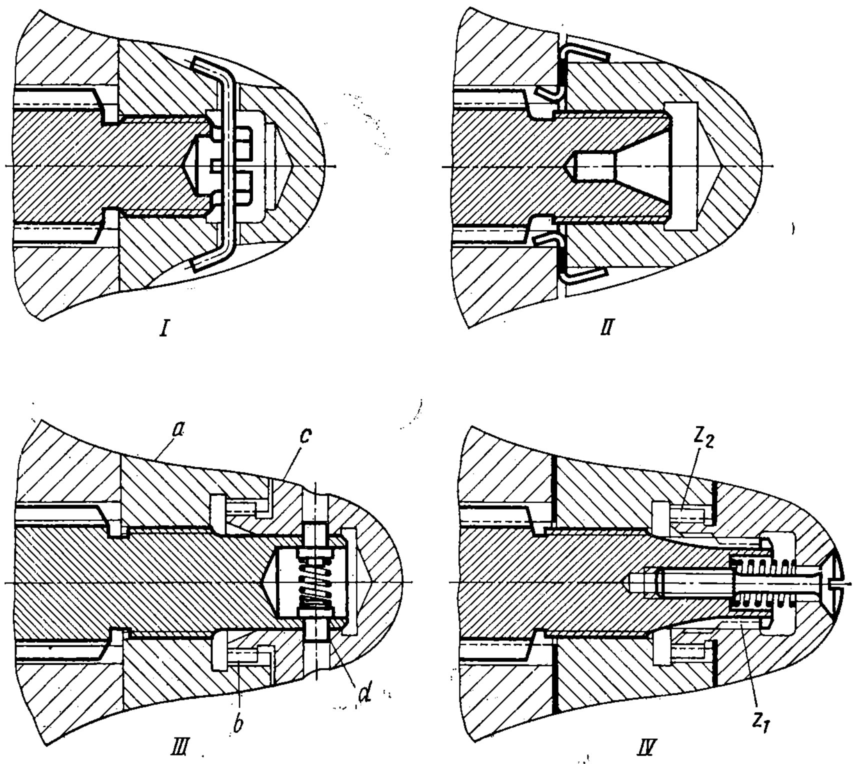


Fig. 706. Methods for locking cap nuts and form nuts on shafts of centrifuge-type machines

Locking by deformable plastic inserts is a frequently used method (see Fig. 705). The head of the threaded plug is fluted, and it is

placed in counterbore with a fluted wall. A ring of metal (usually lead) is forced into the clearance between the head and the counterbore wall. The metal fills the flutes and firmly secures the plug. This method is often used for joints subjected to sealing, e.g. for plugs covering check points in instruments.

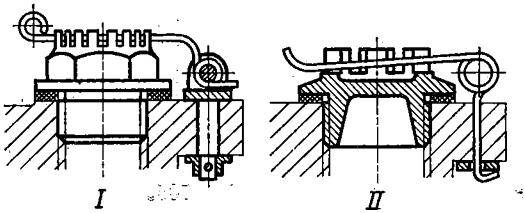


Fig. 707. Locking by swingable springs

Sometimes, a pressurized plastic is injected into the clearance between the plug head and the counterbore wall.

Locking cap nuts and special-form nuts screwed on the ends of shafts in centrifuge-type machines is shown in Fig. 706, *I-IV*. In the assembly shown in Fig. 706, *III*, nut *a* torqued by splines *b* is locked with cap *c*. The cap is set on the plain portion of the shaft

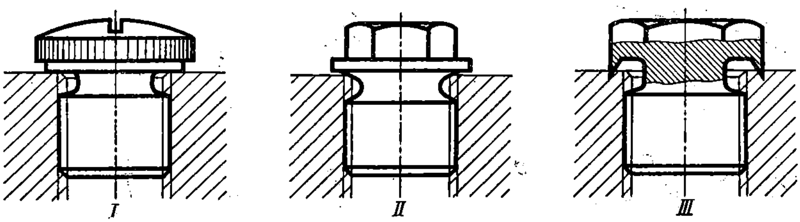


Fig. 708. Threaded plugs set for sealing and permanent locking

end and is fixed by spring-loaded lock pins *d*. The nut can be locked in the angular positions corresponding to the number of the splines.

A method which allows locking in angular positions at small angles to each other is shown in Fig. 706, *IV*. The shaft end is provided with splines whose number is fewer by 1 than the number of splines in the nut. As with verniers, the angle of rotation between adjacent angular positions equals $1/z_1z_2$, where z_1 and z_2 are the numbers of splines on the shaft and the nut, respectively.

Threaded Plugs. Threaded plugs which are to be frequently unscrewed are locked by the methods illustrated in Fig. 707, *I*, *II*. To release the plug for turning, the spring is retracted from the plug slots and is swung aside.

Shown in Fig. 708 are methods whereby threaded plugs are set to seal or to form a permanent joint. When tightened, the plug bears down against the base by the shoulder (Fig. 708, *I*), taper (Fig. 708, *II*) or by sharp circular edge (Fig. 708, *III*). In the latter instance, the base metal is forced towards the plug axis, and it squeezes on the plug thread to seal and lock it securely. The method is used for permanently set threaded plugs.

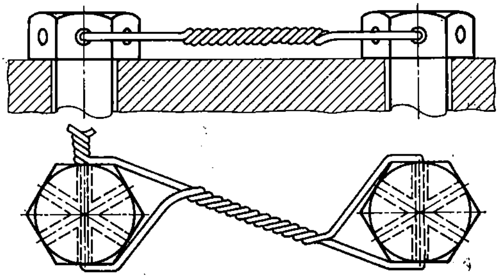


Fig. 709. Wiring bolts in pairs

Locking fasteners arranged in pairs or sets is facilitated by the possibility of using the adjacent fastener to fix the locking device itself. Figs. 709-715 give examples of locking nuts or bolts installed in pairs.

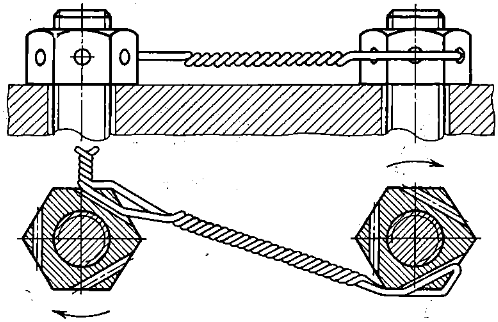


Fig. 710. Wiring nuts in pairs

Wiring is very often used for the purpose, the wire being passed through holes drilled on the flats of hexagon nuts (Fig. 709) or bolt heads (Fig. 710). With castle nuts, the wire is passed through holes in the bolt and slots in the nut (Fig. 711).

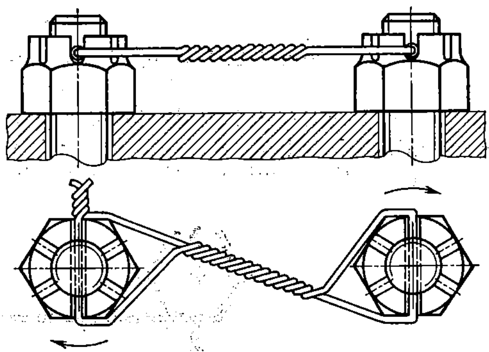


Fig. 711. Wiring castle nuts in pairs

It is important to observe the following rule: tightening the wire by stranding its ends must develop a moment which will tighten the nut. For nuts with a right-hand thread this rule amounts to the requirement that the inclination of the wire between adjacent fasteners should be as shown in Fig. 712, *II* (as viewed from above). The reverse inclination is unacceptable since the moment developed in stranding the wire ends will tend to loosen the nut (Fig. 712, *I*).

The wiring shown in Fig. 713 is also unacceptable, for the wire tension tends to loosen one of the nuts.

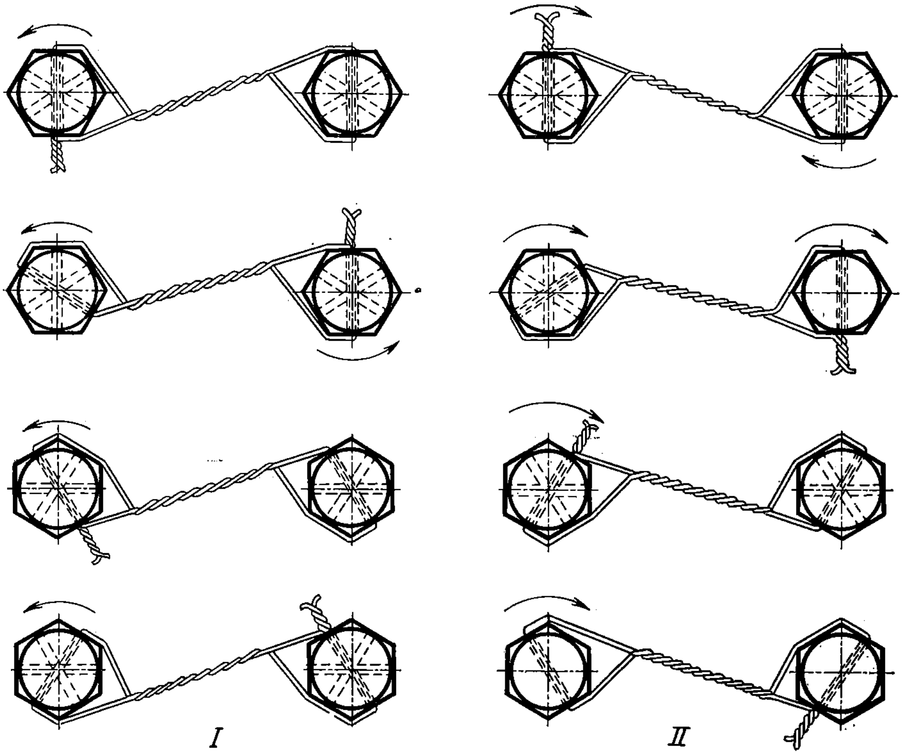


Fig. 712. Wiring of fasteners arranged in pairs
I—incorrect; *II*—correct

The tightened wire slackens in time and the locking becomes unreliable. For this reason, wiring is used for fasteners located close to each other, where the slackening is not considerable.

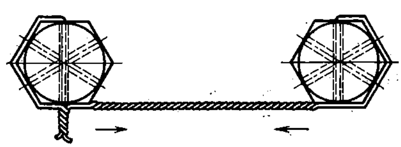


Fig. 713. Improper wiring of a pair of bolts

Locking plates with tabs bent back against the flats of nuts (Fig. 714, *I-IV*) provide more reliable locking. It should be noted

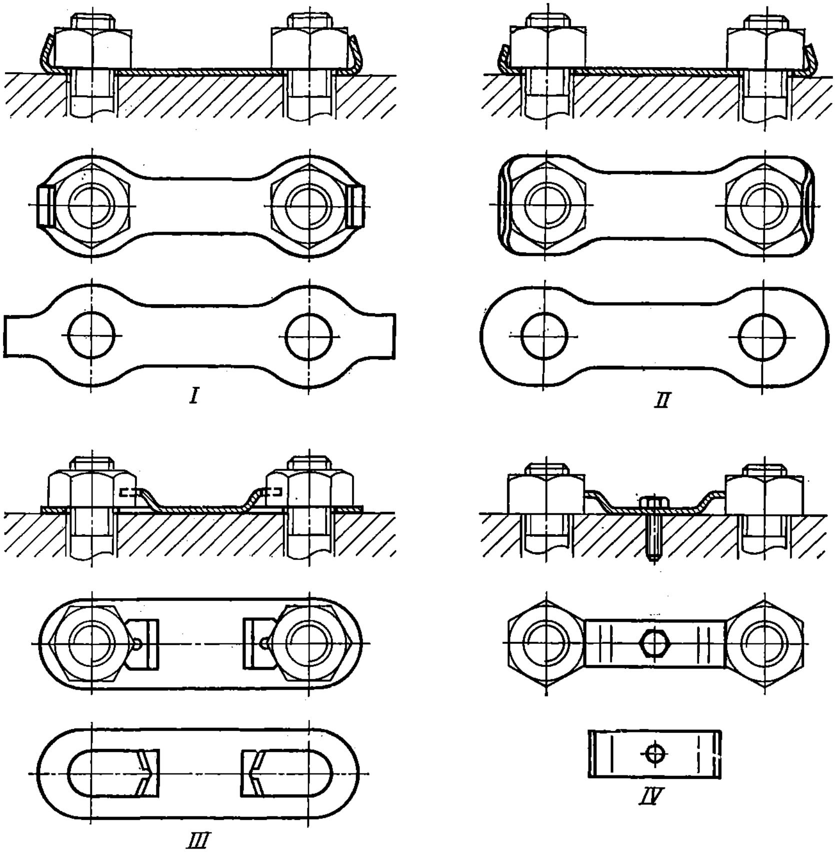


Fig. 714. Nuts in pairs secured by locking plates with tabs

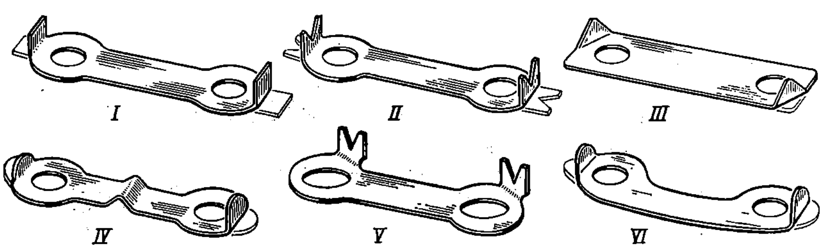


Fig. 715. Locking plates with tabs

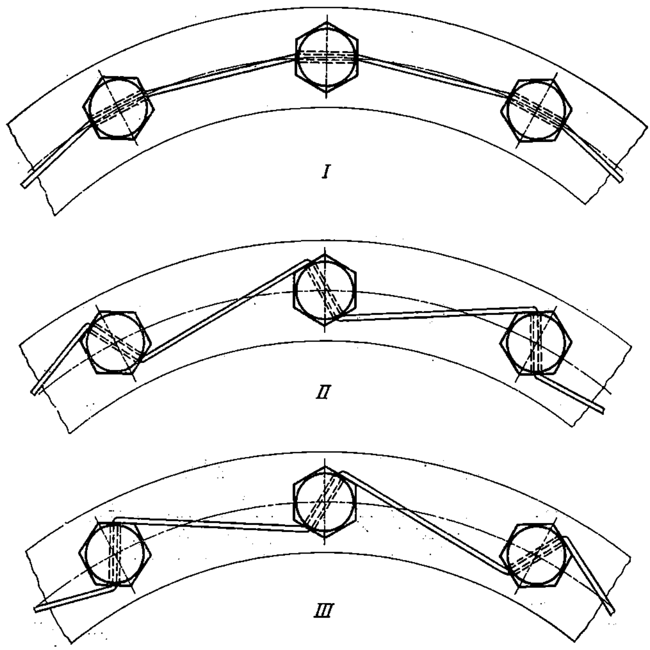


Fig. 716. Wiring of bolts on cylindrical flanges
I, *II*—incorrect; *III*—correct

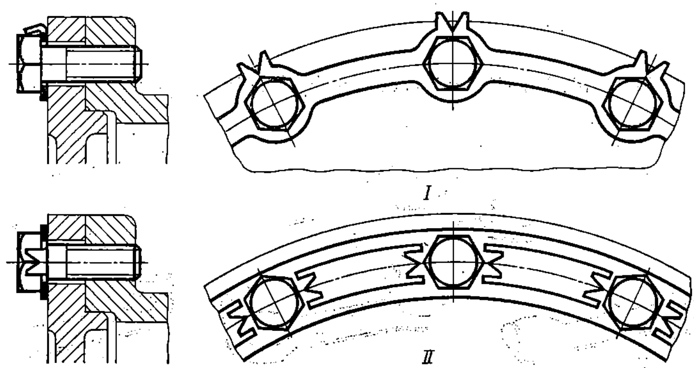


Fig. 717. Locking of cap screws on cylindrical flanges by circular locking plates with tabs

that the arrangement in Fig. 714, *III* gives rise to eccentrical loads on the nuts and therefore can be used only in non-critical applications.

Fig. 715, *I-VI* shows locking plates with tabs in various forms. The plate of Fig. 715, *IV* is provided with a compensating feature which allows the distance between the holes' centres to be adjusted.

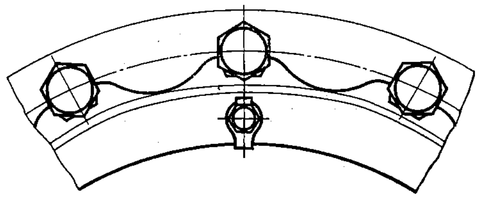


Fig. 718. Locking of bolts on cylindrical flange by circular locking plate recessed for bolt heads

In the plate of Fig. 715, *V*, the same effect is achieved by the oval-shaped holes.

Locking plates are made of soft sheet steel; they are disposable and replaced by new ones after disassembly

The same methods of locking are used for fasteners installed in sets (e.g. bolts on cylindrical flanges): wiring (Fig. 716) and locking plates (Fig. 717, *I* and *II*).

Fig. 718 shows an instance of a set of bolts locked with a screwed-on circular plate with recesses for bolt heads.

8.12. Locking with Paint and Lacquer Coatings

In some indoor applications operating without jarring and impacts, the externally located fasteners are fixed by paint or lacquer coat-

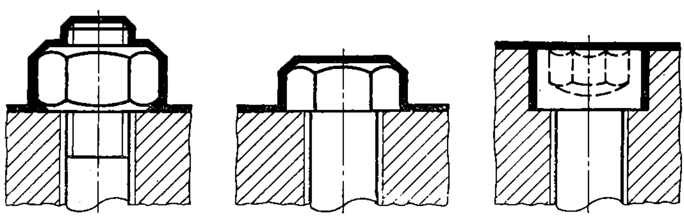


Fig. 719. Locking nuts, bolts and cap screws by coating

ings deposited on the machine surface (Fig. 719, *I-III* and 720, *I-III*) rather than locked with special devices. This method cannot

ensure reliable locking, but nevertheless it prevents fasteners from loosening.

The use of modern coatings based on synthetic resins (e.g. silicone resins) which give a strong elastic film resistant to the environment markedly increases reliability of this locking method.

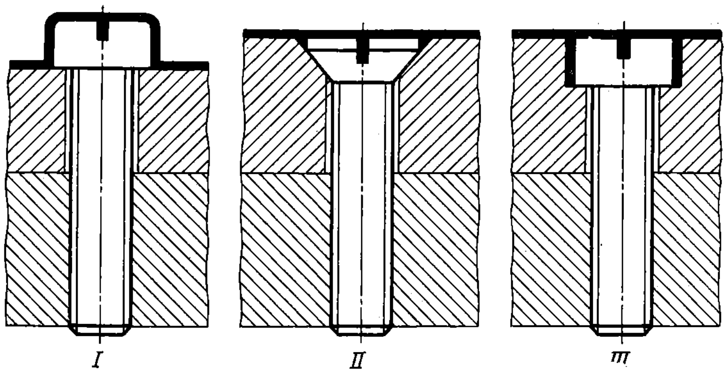


Fig. 720. Locking machine screws by coating

Especially strong adhesion between the fastener being locked and the part it fastens is achieved when the coating material fills the circular clearance between the fastener and the part (Fig. 719, *III* and Fig. 720, *III*) set flush with one another.

Index

Printed in the Union of Soviet Socialist Republics